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Final Report No. 111-2

ULTRASONIC CORPORATION
61 Rogers St., Cambridge 42, Mass.

UNDERWATER ACOUSTIC SOURCE

FOR PERIOD BEGINNING 1 December 1951

ENDING 31 August 1952

SECOND REPORT

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I. Review of Project Objectives and Accomplishments to March 1, 1952

A. Project Objective

The aim of this project was to investigate underwater sound transducers which were primarily mechanical. The following characteristics were desired:

1. Acoustic power output - approximately 1 watt
2. Frequency - 5 to 20 KCS
3. Time duration of output - 8 seconds, minimum
4. Radiation pattern - omni-directional in one plane
5. Size - cross section in plane of radiation should occupy an area within a 4 inch diameter circle
6. Possible power input - that attainable from water-source relative velocity

B. Work Done to March 1, 1952

The first three months were spent on an analytical evaluation of various devices which seemed most likely to meet the desired characteristics, and on the construction and testing of an underwater whistle. The results of tests on this whistle were not favorable so further investigation of it was postponed. (See Ultrasonic Corporation's Technical Progress Report No. 111-1 for Office of Naval Research) The whistle was the only proposed device capable of operating on a water-source relative velocity head. Investigation was then directed toward devices requiring stored energy. It was thought that these devices could better attain the desired performance characteristics. Three such

devices were considered:

1. Driven Diaphragm
2. Water Hammer
3. Hartmann Generator

The first two would operate on pressurized water, the third device on pressurized air. Experimental models of these devices were designed and built. This second report summarizes the work done on the three devices listed above and draws conclusions as to their applicability for use as underwater sound sources.

II. <u>Fluid Driven Diaphragm</u>

A. Explanation of Operation

This device (Assembly Drawing, Figure 1; Photographs, Figures 2 and 3) can operate on fluid supplied at constant flow rate or constant pressure. In general, operation of the device depends on a fluid jet impinging on an edge clamped diaphragm. Motion of the diaphragm reacts on the fluid stream, the interaction resulting in energy being fed into the oscillation of the diaphragm. Such a device was run successfully by Kerr Grant[1] in air, powered with air at constant pressure, and in water, powered by water at constant pressure. Burrell[2] predicts that oscillations can be sustained with an incompressible fluid,

1. Kerr Grant, Proceedings of the Physical Soc. of London, Vol. 34 (1921-1922)

2. Frederick S. Burrell, A Study of a Fluid Driven Acoustic Generator, Mechanical Engineer Thesis at Massachusetts Institute of Technology

supplied at constant flow rate.

B. Design of Device

Operation under water is not practical using air as the driving fluid. The air flow rate required to supply enough energy to cause 1 watt of output is extremely large. Our device was designed to operate on water at a pressure of 400 psi or less. Finishes on parts were made fine and tolerances small to insure perpendicularity between the nozzle and the diaphragm. Fine adjustment of the nozzle-diaphragm distance was provided, for this was known to be critical.

Since the vibrations are self-excited they take place at the natural frequency of the diaphragm. The following diaphragms (edge clamped) were used:

Diaphragm No.	Effective Diameter (inches)	Thickness (inches)	f_n (approximate) (cps)
1	1	.033	7000
2	1	.022	5000
3	1	.012	3000

C. Testing of Device

1. Compressed air driving fluid

A preliminary test was made on the device operating in air and using air supplied at constant pressure. Oscillations were detected by ear but were very faint - too small to measure with the equipment at hand. The frequency was estimated to be about 3 - 4 KCS. Diaphragm No. 1 was used. Vibration was very unstable and extremely sensitive to nozzle-diaphragm distance.

2. Pressurized oil driving fluid

A schematic of the flow system using oil pressurized by a constant displacement pump is shown in Figure 4. This system actually approximates one of constant flow rate since the restriction due to the Fluid Driven Diaphragm varies at a frequency of 3 - 4 KCS which the pump cannot follow. The pump "sees" an average constriction and pumps fluid through the device at a constant flow rate. Diaphragm vibrations were noted by ear with the device surrounded by both air and oil. They were, however, too weak to measure, unstable and very sensitive to nozzle diaphragm adjustment. Frequency was estimated to be about 3 - 4 KCS; Diaphragm No. 1 was used.

3. Pressurized water driving fluid

A schematic of the blow-down flow system used is shown in Figure 5. This system actually supplied water at constant pressure. All three diaphragms were tested, both in air and under water. Vibrations could not be excited. Upstream pressures from 50 - 100 psi were tried. The nozzle diaphragm distance was varied from 0 - .01 inches.

4. Conclusions from tests

The only system which operated when submerged in liquid (oil in this case) was the one supplied with oil at constant flow rate. Even when supplied with liquid at constant flow rate, the device could not be made to operate steadily with a satisfactory power output. This might be explained by:

a. Time changes of nozzle-diaphragm setting

This distance was about .001 - .002 inch. Changes

as small as .0002" would then be a 10% change. Effect on the operation could be appreciable.

b. Variation of flow from that of a radially symmetrical pattern

A basic requisite for this device is that the fluid flow between nozzle and diaphragm be radially symmetrical. Any irregularities in the surface of the nozzle or diaphragm or any departure from nozzle-diaphragm perpendicularity would have an effect on this symmetry. With such a small flow passage between nozzle and diaphragm (.001 - .002 inch), these influences could be strong ones.

D. Conclusions as to Possible Use of This Device

1. System with constant flow rate source

The power output and operating stability of this system were unsatisfactory. The constant flow rate source would be difficult to obtain; it would have to be a constant displacement device such as a vane or piston pump. It would also require a motor or turbine drive. Such a power source would be quite expensive. This version of the Fluid Driven Diaphragm does not seem applicable to the use intended.

2. System with constant pressure source

The system intended for the present application was that utilizing water at constant pressure. The pressurized water source could easily be obtained by using a tank of water with pressurized air above the water, as was done in testing the device. This system failed to operate. Either the

operating point required such critical adjustment of parameters that it was not discovered, or the design itself made operation impossible.

Testing of this device was conducted in the absence of theoretical analysis or comprehensive experimental data. Consideration given to the limited information available at present on this system and to the limited time and funds available under this contract led us to discount the Fluid Driven Diaphragm operating on a constant pressure source as an applicable device.

3. Dimensional requirements

The basic differences between the constant pressure source device under consideration and that operated by Grant[3] are size and frequency. Our smaller, higher frequency device places much more emphasis on dimensional accuracy. Although the best engineering tolerances were used in our design, they were apparently not good enough. Improved dimensional accuracy does not seem practical.

III. Hartmann Generator

A. Explanation of Operation

The operation of this generator surrounded by free air is well documented[4]. Basically the generator consists of two parts: an air nozzle which is operated below its critical pressure ratio,

3. Kerr Grant, op. cit.

4. J. Hartmann, On the Production of Acoustic Waves by Means of an Air Jet of a Velocity Exceeding That of Sound, Phil. Magazine, Series 7, Vol. II, 926-948 (January-June 1931)

so as to set up an oblique shock wave pattern downstream, and a cavity resonator. When the resonator is placed at certain points in the shock wave pattern, strong acoustic vibrations are generated, with frequency determined by the cavity size and power output dependent on the air flow rate from the nozzle and its operating pressure ratio. The air vibrations are caused by the entrance and discharge of air from the cavity resonator, hence the device approximates a velocity source generator.

B. Application to Use Under Water

The device must operate in a medium of air or similar gas. Further, the acoustic impedance into which the device operates must be relatively low to obtain good transmission of power away from the source. When operating into a free field in air, the acoustic impedance apparent to the source is a low one, roughly the characteristic acoustic impedance (ρc) of air. In order to use this device to generate sound under water it must be enclosed in a gaseous medium with provision for transmitting the sound waves into the water surrounding the enclosure. Such enclosure would cause reflections of the sound waves back to the source, affecting the acoustic impedance existent at the source. For good operation of the device, the enclosure should also be designed to cause a low acoustic impedance at the source location.

C. Hartmann Generator No. 1

1. Acoustic design

Radiation, omni-directional in one plane, was desired. A

generated cylindrical wave might be used to approximate this condition. Further, the Hartmann Generator could be designed so that its output would be roughly cylindrical. With this in mind the generator shown in Figure 6 was designed.

The generator itself (nozzle and cavity resonator) is located on the longitudinal center line of a 1 inch radius, steel cylinder. The natural frequency of this cylinder in its first radial mode is about 30 KC. When this cylinder is surrounded by water, the acoustic load apparent to a 30 KC cylindrical sound wave at the cylinder wall would be predominantly one of high resistance (Appendix, Part A). This acoustic impedance at a wall located on a 1 inch radius should result in a low acoustic impedance near the center of the cylinder, which condition would satisfy the generator load characteristic. If the generator were adjusted to operate at 30 KC the steel cylinder would be driven at its natural frequency and should have little reaction on the sound wave. Since the characteristic acoustic impedance (ρc) of the surrounding water is much greater than the characteristic acoustic impedance (ρc) of air an impedance mismatch would exist at the outer surface of the cylinder. Sound energy would be reflected at this interface and a standing wave pattern would be set up inside the cylinder. At the air-cylinder interface there would be a velocity node and a pressure antinode, while at the source (in the center of the cylinder) there would be a velocity antinode and a

pressure node. Hence, the source requisite of high velocity would be met while the large pressure amplitude at the wall would drive the cylinder, forcing sound energy into the water. The standing wave pattern should build up in amplitude until the transmission of energy through the cylinder equals that radiated by the source at the center. This qualitative explanation of operation is based on the theoretical analysis presented in the Appendix, Part B.

2. Mechanical design

In the appendix of Ultrasonic Corporation's Technical Progress Report No. 111-1 for the Office of Naval Research the amount of stored gas necessary to operate this device for 10 seconds was calculated. It was found that about $\frac{1}{4}$ ft^3 of air (or similar gas) stored at 3,000 psi would suffice. The pressure ratio across the Hartmann Generator nozzle should be about $\frac{1}{3}$. $\left(\frac{P\ \text{downstream}}{P\ \text{upstream}} = \frac{1}{3}\right)$ for good operation. By throttling the 3,000 psi gas down to a pressure of 1,200 psi upstream of the nozzle, and by maintaining the downstream pressure inside the steel cylinder at 400 psi this pressure ratio could be maintained. Further, the air inside the cylinder would discharge into the water to a depth of 200 ft. (86.5 psia) at constant flow rate, since the flow would be choked.

The thickness of the resonant cylinder was determined by the strength necessary to withstand an inside pressure of 400 psi. "O" rings at both ends of the cylinder provided a pressure tight seal for the air chamber.

D. Testing of Hartmann Generator No. 1

1. Generator in air

The steel cylinder was left out in this test. In the schematic, Figure 7, the parameters pertinent to evaluating the operation of this generator are shown. The parameters were adjusted as follows:

$$P_{ups} = 50 \text{ psia}$$
$$\delta = 0.167 \text{ inch}$$
$$\Lambda = 4\tfrac{1}{2} \text{ inches}$$

The resonator length (ℓ) was varied from $1\frac{1}{2}$ inches to less than $\frac{1}{32}$ inch and the frequency and pressure amplitude were measured. The frequency varied with ℓ as shown in Figure 8; pressure amplitude varied with ℓ as shown in Figure 9. The power output computed for a RMS pressure amplitude of $6{,}000\ \frac{\text{dyne}}{\text{cm}^2}$ ($f \approx 30$ KCS) is roughly 45 watts[5]. This assumes a cylindrical wave shape of height about 3 inches. (This height was estimated by moving the microphone vertically 1 or 2 inches above and below the horizontal center line and observing little variation in pressure amplitude.) The waveform of the pressure variation, as indicated by the microphone, was high in harmonic content at the lower frequencies. At a frequency of about 10 KC and up the wave became more purely sinusoidal until at 30 KC it was nearly

5. This estimate is only intended to establish order of magnitude. A sample of the method of calculation used is included in the Appendix, Part C.

a perfect sinusoid.

These measurements were performed to provide a check on the frequency and output characteristics of the Hartmann Generator without the enclosure. The upstream pressure of 50 psia was used so that an available 125 psi constant pressure supply would suffice. Since the downstream pressure was 1 atm., the pressure ratio across the nozzle was about 3 to 1. This ratio is the same as that intended at a higher upstream pressure (1,200 psi) so that the effect of using the 1,200 psi upstream pressure should be an increase in power output of the generator with no other appreciable change in operation.

2. Generator in water

With the steel cylinder in place the device was placed under water in the anechoic tank (Figures 10 and 11). With parameters adjusted as follows:

$$P_{ups} = 50 \text{ psia}$$

$$\delta = 0.167$$

$$\Lambda = 2 \text{ inches}$$

ℓ was varied and pressure amplitude and frequency were observed. The output showed pronounced maximums at about 9 and 16 KCS, but even at these points it was much lower than expected. The best operating point was at a frequency of 9 KCS. The pressure amplitude was about 3,000 dyne/cm^2 corresponding to a power output of roughly 1.4×10^{-3} watts[6]

6. This estimate is only intended to establish order of magnitude. A sample of the method of calculation used is included in the Appendix, Part C.

(again assuming a cylindrical wave of height, 3 inches).

3. Conclusions from test data

a. Steel cylinder

This cylinder, resonant at about 30 KC, was not being driven at its natural frequency. It is also very probable that the cylinder was highly damped due to its mounting.

b. In order for energy to be transmitted to the water it is necessary that the acoustic conditions previously described exist. Any deviation of the sound field inside the steel cylinder from that of cylindrical standing waves would seriously reduce the acoustic power radiated into the water. In the Hartmann Generator No. 1 the longitudinal distance from the source on the longitudinal centerline of the cylinder to the end of the air chamber is appreciable with respect to a wavelength (distance is 1 inch while for $f = 30$ KC, $\lambda = 0.44$ inch). It is very possible that any energy radiated longitudinally from the source at the center would be reflected causing the sound field inside the cylinder to differ from that of a cylindrical standing wave pattern. That energy was being radiated longitudinally from the source at the center was verified by a few measurements which indicated appreciable pressure amplitudes at points relatively distant from the source in the longitudinal direction.

4. Modification of Hartmann Generator No. 1

a. Aluminum spacers

To assure a cylindrical sound field inside the steel cylinder, two aluminum spacers were inserted as shown in Figure 12. By cutting down the longitudinal dimension of the sound passage the effect of energy reflected from the ends should be greatly decreased. A space between the O.D. of the spacers and the I.D. of the steel cylinder serves as the passage for air discharge.

b. Effect of spacers on operation of generator

When tested underwater the output of the modified device did not differ appreciably from that of the device without the spacers. The placement of the air discharge passage at the inner wall was dictated by mechanical practicability. From an acoustic standpoint, however, it would have been better to avoid a sound energy escape path at a pressure antinode in the sound field. The position of the discharge passage at the inner wall of the cylinder made it difficult, if not impossible, to sustain a large pressure amplitude at that point.

E. Hartmann Generator No. 2

1. Difficulties encountered in Generator No. 1

Three points were derived from tests on Generator No. 1:

a. A steel cylinder with air tight seals was difficult to excite to vibration.

b. The longitudinal dimension inside the cylinder could have a strong effect on operation.

c. The air discharge opening should be placed at some point of low pressure amplitude.

2. Design of Generator No. 2

The generator shown in Figures 13, 14 and 15 was designed as an improvement over Generator No. 1. It was expected to operate best at about 9 or 16 KCS as predicted by the analysis in the Appendix, Part B, and as demonstrated by tests on Generator No. 1. The steel enclosing cylinder was replaced with an annular ring of medium soft, commercial rubber. (Characteristic acoustic impedance of this material, ρc, close to that of water.) The longitudinal dimension of the air chamber was kept small. The air escape passage was placed at a point in the sound field where it would be near a pressure node at either 9 or 16 KCS. (Roughly a quarter wavelength toward the center, from the rubber-air interface.) It was expected that this design would result in a well defined cylindrical standing wave field in the enclosure.

F. Testing of Hartmann Generator No. 2

1. Generator in air

This test was conducted without the rubber ring. A schematic of Generator No. 2 is shown in Figure 16. The parameters were adjusted as follows:

$$P_{ups} = 50 \text{ psi}$$

$$\delta = 0.167 \text{ inches}$$

$$r = 1\tfrac{1}{2} \text{ inches}$$

$$y = 0.3 \text{ inches}$$

The parameter ℓ was varied and frequency and pressure amplitude were noted. This device exhibited much the same characteristics as Generator No. 1 when operated in an air medium (Figures 8 and 9).

2. Generator in water

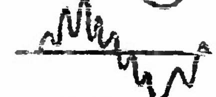
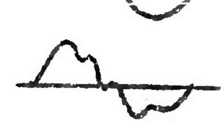
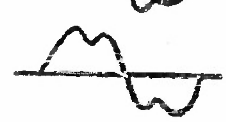
With the rubber enclosing ring in place, the device was placed underwater in the anechoic tank. The parameters were adjusted as follows:

$$\delta = 0.164 \text{ inches}$$

$$n_c = 10\frac{3}{2} \text{ inches}$$

$$y = 0.187 \text{ inches}$$

At three different upstream pressures, the downstream or chamber pressure and the frequency were varied to find the best operating points (i.e., maximum pressure amplitude). The chamber pressure was controlled by a valve in the discharge line while frequency was controlled by adjusting ℓ, the cavity length. The following was observed:

Upstream Pressure P_{ups} (Psig)	Chamber Pressure P_{cham} (Psig)	Cavity Length ℓ (Ins)	Frequency f (KCS)	Pressure Amplitude P_{rms} (dyne/cm^2)	Wave Form of Pressure Variation
200	60	$\frac{[illegible]}{16}$	16.8	3200	
300	60	$\frac{5}{16}$	9.2	3500	
300	100	$\frac{3}{16}$	16	2400	
600	100	$\frac{5}{16}$	9	9500	
600	100	$\frac{3}{16}$	15.5	4700	

The chamber pressure was in all cases adjusted to give the maximum power output. When reduced below values tabulated the output was not noticeably affected.

Various nozzle-resonator distances (δ) had been tried before this test was run, to evaluate the importance of this parameter on operation. Variation of this dimension of .03 inches from its value at a good operating point resulted in a decrease in power output by roughly a factor of two. The order of magnitude of the output of the generator, then, was not seriously affected by small variations in this adjustment.

The radiation pattern observed by measuring pressure amplitude at various points around the generator seemed to be more spherical than cylindrical. This was surprising, but the pressure amplitudes measured near the outside surfaces of the nozzle and cavity housings at positions 1 and 2, Figure 16, were of the same magnitude as that measured at position 3. Apparently the nozzle and cavity housing pieces were excited to vibration by the standing wave pattern in the air enclosure, causing energy to be radiated longitudinally. This phenomena was not observed in the tests run without the rubber enclosing ring, at which time the radiation pattern appeared to be predominantly cylindrical.

The pressures needed to operate this device were supplied by carriages of twelve large gas bottles containing nitrogen initially at 2,000 psi. The testing time provided by this pressure source was very limited. (Roughly 10 minutes for

constant operation with P_{ups} = 1,200 psi.) To lengthen the time available for measurements, it was decided to test the generator at pressures lower than 1,200 psi, hence the previous data. It is reasonable to assume that results obtained at pressures below 1,200 psi can be extrapolated to indicate order of magnitude of power output at a 1,200 psi upstream pressure.

3. Conclusions from tests

Maximum outputs were observed at 9 and 16 KCS as predicted by the analysis in the Appendix, Part B. With an upstream pressure of 600 psi and frequency adjusted to 9 KC a steady wave with pressure amplitude of 9,500 dyne/cm^2, measured at a point $10\frac{1}{2}$ inches from the source, was generated. If a spherical radiation pattern is assumed this pressure amplitude corresponds to approximately 0.52 watts of radiated power. Presumably, if the upstream pressure were doubled to 1,200 psi and the same pressure ratio across the nozzle were maintained, the air flow rate through the device would double and the power output would double.

The chamber pressure when adjusted below a certain value (those tabulated, for example) had no noticeable effect on output. The data taken seems to indicate that the pressure ratio across the nozzle for best operation is greater than the 3 to 1 ratio which was expected to be best.

C. Conclusions as to Possible Use of the Hartmann Generator

Our final model of this device (Generator No. 2, Figures 13, 14 and 15) does not possess all of the characteristics considered

to be desirable for the underwater sound source in question.

Its shortcomings are:

1. It requires stored energy
2. Its power output is only $\frac{1}{2}$ watt as compared to the desired value of 1 watt
3. Its radiation pattern, which seems to be predominantly spherical, causes the power to be spread over a larger area than it would be in a cylindrical sound field. This decreases the intensities measured at points in the horizontal plane of the device.

Though stored energy (in the form of compressed gas) cannot be avoided, the power output and radiation pattern of the device can be further influenced. A design which minimized the vibration of the nozzle and cavity housing pieces would decrease the power radiated longitudinally. It seems probable that a corresponding increase in power radiated in the horizontal plane of the generator would result. An increase in the thickness of the housing pieces might well accomplish this.

The size of the resonant cavity used in the source at its center is related to operation of the source in that the length of the cavity, $\hat{\ell}$, determines the frequency of operation and the diameter of this cavity must be small with respect to a wavelength at any frequency to prevent cross modes of air vibration. Further, it is likely that if the diameter of the cavity resonator is increased within this

limitation, the efficiency of the device will be improved, resulting in more power output for a given input. An operating frequency as high as 30 KC was originally contemplated and at such a frequency the wavelength in air is about 0.44 inches. The length of a cavity necessary to produce this frequency is a quarter wavelength or 0.11 inches. The recommended diameter for a Hartmann Generator is given by:[7]

$$l > d > \frac{l}{1.5}$$

d = cavity diameter

l = cavity length

In this case values of d between 0.11 inches and 0.073 inches satisfy this condition. The value of 0.093 inches used, was chosen as an average between these two. Further, this value is the same as the diameter of the nozzle, a feature which simplifies alignment of nozzle and cavity. At the lower operating frequencies (9 and 16 KCS) found to be desirable from our tests, the cavity diameter could be as large as 0.20 inches and still be within the range of recommended values.

It is impossible to predict how much the power output for a given input would be affected by increasing the cavity diameter. This would have to be determined by experiment. But since our Generator No. 2 does not have as large a

7. Van L. Ehnet and H. Hahnemann, Ein Neuer Schall und Ultraschallgeber zur Erzeugung Starker Intensitaten in Gasen. Feitschkrift fur Techinsche Physik 23 Jahrg. 1942, pp. 245-266 (Printed by Edward Bros., Inc., Ann Arbor, Mich.)

value of cavity diameter as that recommended, increasing this dimension should have a favorable effect on the power output.

In conclusion, the Hartmann Generator for use under water, in its present stage of development, does not completely satisfy the requirements imposed on the underwater acoustic source of interest in this contract. It does seem possible, however, to influence the operating characteristics of the device to more nearly fulfill these requirements.

IV. Water Hammer Device

A. Explanation of Operation

This device, shown in Figures 17 and 18, is partially understood[8]. If a valve, which tends to close and decrease the flow through it when the pressure upstream is increased, is placed in a pipe line containing pressurised fluid, a phenomenon called "water hammer" can take place. The device drawn in Figure 18 utilises this phenomenon to produce sustained vibrations. Its operation can best be explained in four parts:

1. Initiation of vibrations

With flow passing through the valve, a small displacement of the valve toward the closed position would decrease the flow through the valve causing an increase in pressure adjacent to the valve. This pressure increase would be

8. Sumiji Fujii, A Self Induced Water Hammer, Science of Machine (Kikai No Kenkyu) Vol. 1, No. 11 (Nov. 1949)

propagated upstream as a pressure wavefront. At a distance, L, upstream from the valve there is a sudden increase in area in the cross section of the tube. This "open" termination will cause the compression wavefront traveling upstream to be reflected as a pressure decrease (rarefaction) wavefront. Practically all of the energy contained in the wave is reflected by this termination so that the pressure amplitudes of the incident and reflected waves are essentially the same. The rarefaction wavefront now will travel downstream to the valve. Here two things occur:

a. Under the influence of this pressure decrease wave the valve will open increasing the flow and causing a second pulse (rarefaction) to propagate upstream.

b. The original rarefaction wave will be reflected from the valve (a stiff termination), remaining a rarefaction wave, and will propagate upstream. If we assume that no energy is taken from this original wave by valve motion, the incident and reflected waves again have the same amplitude.

The two waves now traveling upstream will add and the resultant wave will be larger than the one which a moment before had been traveling downstream, the difference being due to the change in flow caused by valve motion. This rarefaction wave will now travel upstream to the open termination, be reflected as a compression wave and return to the valve. Now the valve will close slightly, the compression wave will be reflected as a compression wave while its amplitude is

increased by the decrease of flow through the valve. The pressure wave will now propagate upstream. At this time, after 4 traverses of the tube length, L, the disturbance is again traveling upstream as a compression wave, and the cycle of events just described will be repeated. Each time a wavefront impinges on the valve it will be reflected and its amplitude will be increased as described.

The valve is now excited by alternate compression and rarefaction waves. The time between compression waves is

$$T = \frac{4L}{C}$$

T = period of pulsations

L = length of tube from valve to open termination

C = velocity of propagation of sound in fluid-filled tube

Since $T = \frac{1}{f}$ $\qquad$ f = frequency of pulsations

$$L = \frac{C}{4f}$$

Substituting $f = \frac{C}{\lambda}$ $\qquad$ λ = wavelength of pulsations

The relationship $L = \frac{\lambda}{4}$

is obtained. Hence the length of the pipe must be a quarter wave length. This would cause all generated pulses to be exactly in phase. In the succeeding explanation certain phase relationships, necessary to the operation of the device, will be described which necessitate that the tube length be somewhat less then a quarter wave length.

2. Limitations to build-up of the waves

The assumption that no energy was taken from a wave incident

on the valve by valve motion was made only to simplify the explanation of initiation of vibrations. In reality each wave incident upon the valve can supply energy to any system which might appear as a damping load coupled to the valve. In the present design of the water hammer device, the valve is mounted at the center of a diaphragm so that motion of the valve causes motion of the diaphragm from which a certain amount of acoustic energy is radiated and a certain amount of energy is dissipated by mechanical damping. Energy is also taken from the pressure pulse by other means. Reflection at the "open" termination end of the tube is accompanied by some energy loss. The wave, in traveling up and down the tube, loses energy to the fluid and to the metal tube because of viscous damping.

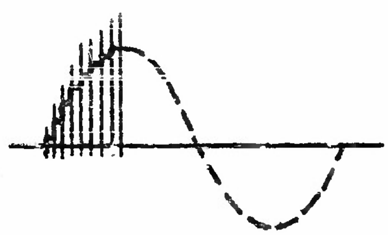
If the amplitude of the rarefaction wave reaches the value of static pressure in the tube, liquid cavitation can occur. Such cavitation would prevent further increase of the pressure wave amplitude and would cause considerable dissipation of energy in heat and turbulence.

Consequently, the pressure amplitude built up by the pressure wave is limited. Since the energy losses mentioned are all directly dependent on wave amplitude, this amplitude will build up until the energy lost from the wave during its cycle (4 traverses of the tube) is just equal to the energy supplied to the wave from flow modulation at the valve.

3. Steady state operation

The simple picture of the traveling compression or rarefaction

wavefront is modified by the physical characteristics of the system. The valve is mounted on a diaphragm, and this unit resembles a damped, spring-mass vibrating system. Though this system is initially excited by a pulse it will vibrate steadily with the sinusoidal motion characteristic of such systems. The pressure variation in the fluid upstream of the valve will also become sinusoidal. Despite the fact that the "simple wavefront" picture does not persist, it is helpful in visualizing the actions involved. These actions remain essentially the same even though the form of pressure variation changes from a pulse to a sinusoid. If one thinks of a sine wave as a series of simple wavefronts as shown below,



the foregoing explanation can be extended to sinusoidal variations.

4. Conditions necessary for operation

a. Phase relationships

For waves to be reinforced when reflected from the valve, the vector representing sinusoidal variation of pressure force at the valve must have a component in phase with the valve displacement vector. This stipulation makes possible the condition that an increase in pressure causes a decrease in flow and vice versa.

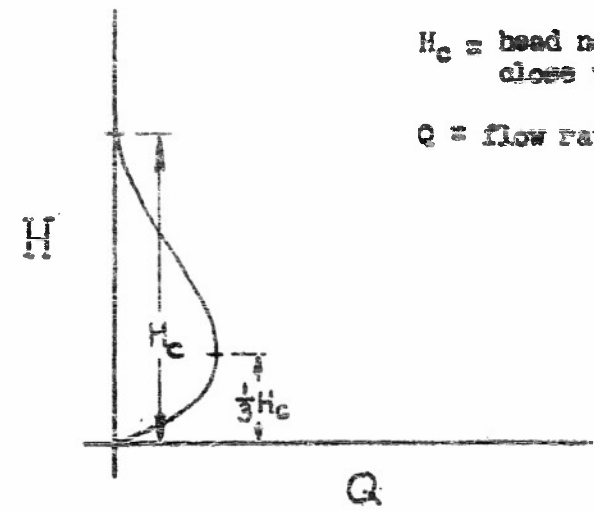
For this phase relationship to be satisfied, the natural frequency of the valve-diaphragm vibrating system must be higher than the frequency of operation so that the spring reactive force on the valve is predominant over the mass reactive force. Concurrently, the reflections from the "open" termination end of the pipe must cause the waves to arrive back at the valve so that the resultant pressure force at that point has a component in phase with valve displacement. The theory of acoustic impedance tubes predicts that the tube length must be less than a quarter wave-length ($L < \frac{\lambda}{4}$) for this phase relationship to exist.

The pressure force vector must also have a component in phase with valve velocity. This component balances the damping forces and determines the power output through the valve-diaphragm unit to mechanical damping of the diaphragm and acoustic radiation. The effect of flow modulation on pressure variation at the valve provides this component.

b. Valve flow characteristics

The flow characteristic for a spring mounted valve which closes linearly with increasing upstream head (pressure) is shown on the next page[9].

9. Sumiji Fujii and Oishiaki Kiyama, Excitation of Vibration of a Fluid Column by a Fluttering Valve, University of Tokyo



H = upstream head (pressure)

H_c = head necessary to close valve

Q = flow rate through valve

This graph indicates that the flow decreases with an increase in head (pressure), only when the head is greater than $\frac{1}{3}$ H_c (H_c = head necessary to close the valve). Consequently, a minimum upstream pressure is necessary to excite vibrations.

The closure head (H_c) is dependent on the initial valve opening (distance between valve and valve seat), the smaller the initial valve opening, the smaller the closure head and vice versa. The minimum upstream pressure necessary to excite small vibrations is similarly dependent on the initial valve opening.

B. Design of Device

Several parts used for the Fluid Driven Diaphragm were used in this device also. The "heart" of this device is the valve-diaphragm unit. Tolerances which affected relative position of valve and valve seat (Figure 17) were closely held to insure

alignment. Fine adjustment of the valve opening (distance from valve to seat in the neutral position) was provided, since this was critical.

The diaphragms used were the same as those used for the Fluid Driven Diaphragm except for the valve mounting hole in the center. However, the natural frequency of each of these diaphragms with the valve mounted at the center was considerably lower.

The sudden expansion of cross sectional area apparent to the sound waves traveling upstream from the valve, usually takes the form of a tank with cross sectional area considerably larger than that of the tube immediately upstream from the valve. The tank used in this experimental design was a high pressure two inch plain pipe coupling with a reducing bushing in each end (Figure 18). The tube from the valve to the area expansion upstream included a piece of $\frac{3}{8}$ inch pipe. This made it possible to vary the distance from valve to area expansion by varying the length of pipe used.

C. Initial Testing of Water Hammer Device

1. Preliminary test of operation

A schematic of the Water Hammer Device is shown in Figure 19. The water flow system used for this device was the same as that used for the Fluid Driven Diaphragm, Figure 5. With Diaphragm No. 1, two values of L were used and the valve opening was adjusted to the point of best operation. With

$$L = 5 \text{ inches}$$

$$P_{ups} = 400 \text{ psi}$$

the following was observed:

Tube length L (Ins.)	Frequency f (KCS)	Pressure Amplitude $P_{max} \times 10^3$ (dyne/cm^2)
2.7	3.9	10 - 13
3.2	3.5	15 - 30

Bubbling was noted at the center of the diaphragm. The outputs were very erratic and waveforms indicated high harmonic content. The better operating condition was with L = 3.2 inches, so with this value established the next test was run.

2. Effects of varying upstream pressure, P_{ups}

With Diaphragm No. 1 in use and with

$\lambda = 5$ inches

L = 3.2 inches

the valve opening was adjusted for best operation over the range of pressures. The following was observed:

Upstream Pressure P_{ups} (psig)	Frequency f (KCS)	Pressure Amplitude $P_{max} \times 10^3$ (dyne/cm^2)	Waveform
200	No vibration	---	---
250	3.7	4.5 - 6.5	
300	3.7	9 - 11	"
350	3.7	9 - 11	"
400	3.7	9 - 11	"
450	3.7	9 - 11	"
500	3.7	9 - 11	"

Bubbling was again noted at the center of the diaphragm. The values of pressure amplitude recorded correspond to the maximum peak to peak value, as shown below:

The frequency recorded corresponds to the frequency at which the maximum peak to peak values occur. The maximum upstream pressure used (500 psi) was limited by the strength of the diaphragm. At the conclusion of this test it was found that Diaphragm No. 1 was badly cracked.

A similar test was run using Diaphragm No. 2. With

π = 5 inches

L = 3.2 inches

the valve opening was adjusted for best operation over the range of pressures and the following was observed:

Upstream Pressure P_{ups} (psig)	Frequency f (KCS)	Pressure Amplitude $P_{max} \times 10^3$ (dyne/cm^2)	Waveform
150	No vibration	---	---
200	2.10	75	
250	2.15	27	"
300	2.18	15	"

Bubbling was again noted at the diaphragm center. Again, the values of pressure amplitude recorded correspond to the maximum peak to peak values while the frequencies recorded correspond to the frequencies at which the maximum peak to

peak values occur. The maximum upstream pressure used (300 psi) was limited by the strength of the diaphragm. Operation of the device was fairly stable with time.

This test was also attempted with Diaphragm No. 3. The diaphragm cracked badly after only a few seconds of vibration and further use of it was impossible.

3. Effects of varying tube length, L

With Diaphragm No. 2 in use and with

$$\ell = 9 \text{ inches}$$

$$P_{ups} = 300 \text{ psig}$$

the following was observed:

Tube length L (Ins.)	Frequency f (KCS)	Pressure Amplitude $P_{max} \times 10^3$ (dyne/cm^2)
6.6	1.7	12
6	1.9	10
5.1	2.1	10
4	2.5	10

At each value of L, the valve opening was adjusted for best operation. Bubbling was noted at the center of the diaphragm. The values of pressure amplitude and frequency recorded have the same significance as in the previous tests of the device.

D. Conclusions from Initial Testing

1. Effect ofupstream pressure, P_{ups}

A certain minimum upstream pressure was necessary, in each case, to start vibrations, its value dependent on the valve opening. The smaller the valve opening the smaller the

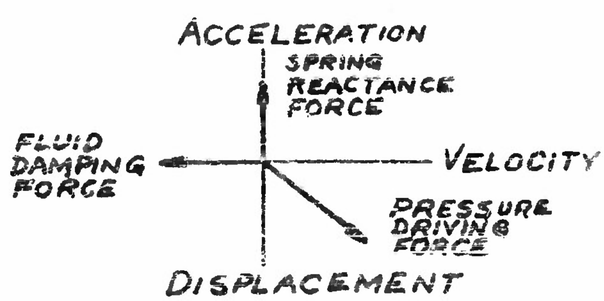
minimum pressure. This fact agrees with the information presented under "Explanation of Operation". An optimum value of upstream pressure seemed to exist for each diaphragm. If the pressure were increased above this value the amplitude of output remained the same or decreased. Further increase beyond this optimum value also seemed to increase the harmonic content of the wave.

2. Effect of tube length, L

Variation of tube length seemed to have little effect on amplitude of output. Its effect on frequency can best be seen by the plot of Frequency versus Tube Length (Figure 20), made from the tabulated data. A curve of $\frac{\lambda}{4}$ (quarter wave length) versus frequency is included for comparison. L is less than $\frac{\lambda}{4}$ at all frequencies measured. This indicates that the spring reactance force due to the diaphragm is predominant over the mass reactance forces acting on the valve at the valve-diaphragm interface. For the resultant reactive force and the resistive damping force to be balanced by the driving force, it is necessary that

$$L < \frac{\lambda}{4}$$

a condition which causes the fluid pressure driving force to lag the velocity of the valve. The force balance, as shown in the vector diagram below, is then possible.



These observations are also in accord with the remarks made previously in "Explanation of Operation."

The small change in amplitude of vibrations with frequency indicates that the diaphragm was highly damped.

3. Effect of using different diaphragms

The test data indicates much higher output amplitudes when Diaphragm No. 2 was used. The stability of operation also seems much better with this diaphragm in use. The lower frequency of operation with Diaphragm No. 2 as compared to Diaphragm No. 1 is less desirable but the other improvements in operation were of such value that Diaphragm No. 2 was considered to be more satisfactory and was used for all succeeding measurements.

4. Cavitation and air bubbles

During the previous measurements, violent bubbling was noted in the water medium surrounding the device at the center of the diaphragm. This could have been caused by liquid cavitation or by air bubbles. In either case, the reflection and dispersion of radiated sound, by such bubbles, was very undesirable.

To avoid the difficulties accompanying formation of bubbles the Water Hammer Device was modified as described in the next section.

E. Modification of Water Hammer Device

To avoid bubbling in the surrounding medium next to the center of the diaphragm, a chamber for containing de-gassed castor oil in contact with the diaphragm was added to the device. The oil

was separated from the surrounding medium by a thin pliofilm membrane. The device with this modification (shown in Figures 21, 22 and 23) represents its latest stage of development.

The use of de-gassed castor oil eliminates air bubble formation in the oil and increases the pressure amplitude necessary to cause liquid cavitation to several atmospheres. The bubbling at the diaphragm-castor oil interface can be avoided in this way. The sound energy radiated through the castor oil spreads out hemispherically so that the intensity at the castor oil-water interface is reduced below the value necessary to cause bubbling. It was found necessary to fill the cavity adjacent to the diaphragm with rubber to prevent water from leaking through the valve mounting screw hole into the castor oil.

F. Testing of Modified Water Hammer Device

The tests already described indicated the effects of the important parameters on operation. To accurately determine the power output, frequency and stability of operation of the device, as modified, the following test was run.

With $L = 4$ inches

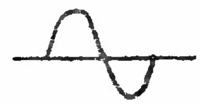
$P_{ups} = 300$ psig

$\Pi = 9$ inches

the device (as shown in Figure 23) was run for a total time of about 20 minutes. The frequency of operation remained constant at about 2.3 KCS. However, the pressure amplitude and harmonic content of the wave varied considerably.

With the valve opening adjusted to give maximum initial output at the established upstream pressure (300 psig) the pressure

amplitude (P_{max}) of the output varied from roughly 4200 to 17,000 dyne/cm^2. The variation took place slowly with time, the amplitude increasing and decreasing about the most constant value of 10,000 dyne/cm^2. The waveform of pressure variation varied as tabulated below:

Frequency f (KCS)	Pressure Amplitude P_{max} (dyne/cm^2)	Waveform
2.3	17,000	
2.3	10,000	
2.3	4,200	

The water flow rate through the device was measured several times during this test and was found to vary from 1.8 to 3 in^3/sec.

G. Conclusions from Testing of Modified Water Hammer Device

The power output from the water hammer device corresponding to the pressure amplitude of 4,200 dyne/cm^2 at a distance of 9 inches from the diaphragm is about 0.02 watts. That corresponding to 17,000 dyne/cm^2 is 0.3 watts. These estimates assume a hemispherical radiation pattern with its center at the diaphragm center. Such a radiation pattern was indicated by traverses about the source with a microphone which revealed that the pressure amplitude was roughly constant at a fixed radius.

The time variation of power output is not easily explained. The most obvious cause for such slow time variations is variation of dimensions of the device due to temperature change

and vibratory creep. The mean value of valve opening (distance from valve to seat) was of the order of .001 inches so that a variation of this dimension as small as .0001 inches (a 10 per cent change) might have a noticeable effect on operation. The diaphragm (No. 2) used for this test was later found to be cracked. It is not known when, during the test, the crack occurred, so correlation between this occurrence and variations in output is impossible.

Our main conclusions from this test were that:

1. A power output of about 0.02 watts (this was the minimum output measured during the time variations) could readily be obtained from this device. Minor changes in its adjustment, which were impossible to control, could increase the magnitude of the output to 0.3 watts.
2. The amount of water required to operate the device for 10 seconds would be about 30 in^3. This water would have to be available at a pressure of 200 - 300 psi.

H. Conclusions as to Possible Use of This Device

Our final model of this device (Figures 18, 21 and 23) does not have several of the characteristics considered to be desirable for the underwater sound source in question. The device exhibits the following limitations:

1. It requires stored energy.
2. Its power output is from 0.02 to 0.3 watts as compared to a desired value of 1 watt.
3. Its best frequency of operation is about 2.5 KCS which is below the desired 5 to 20 KC range.

4. Its radiation pattern is roughly hemispherical as compared to the desired omni-directional radiation in one plane.

As in the Hartmann Generator No. 2 the stored energy (in the form of pressurizedwater) cannot readily be avoided. A motor or turbine driven pump might be used to draw water from the surroundings and pressurize it, but this scheme would be much more complicated, and correspondingly more expensive, than simply storing pressurized water. The storage system visualized for this device is the same as that for the Fluid Driven Diaphragm. The small amount of water necessary for operation for 10 seconds (30 in^3) could easily be stored under an air pressure head of the desired amount.

The operating frequency of about 2 to 2.5 KCS seems to be somewhat of an upper limit for this device. The amplitude of vibrations necessary for a given power output decreases as frequency increases. The stability of operation of the device definitely seems to be a function of the vibration amplitude of the valve-diaphragm unit: the larger the amplitude, the more stable the operation. At frequencies much above 2.5 KCS the stability of the device does not seem good enough for this application. Consequently, operating frequencies above 2.5 KCS do not seem advisable.

The power output of this device can undoubtedly be increased by increasing its size, principally its diameter. If the diameter of the diaphragm were increased, its radiation characteristics would be improved. For the same amplitude of vibrations as the present diaphragm the acoustic resistive load,

into which the diaphragm would work, would appear higher and more energy could be transmitted into the water. In addition, the diameter of the valve could be made larger which would increase the flow through the device. This would result in an increase in power input and a corresponding increase in power output.

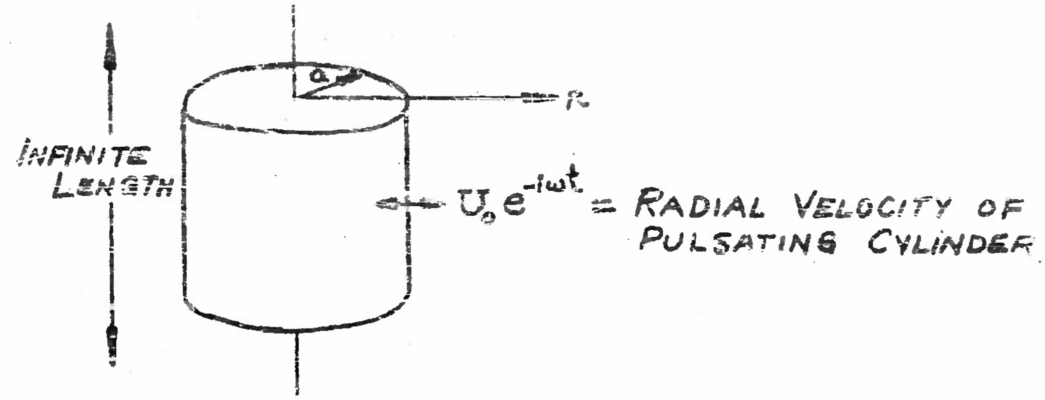
The radiation pattern of the present water hammer device could best be modified to approximate an omni-directional pattern in one plane by a curved horn arrangement. This is a design problem which we have not investigated but its solution does not appear to be difficult.

In conclusion, the water hammer device now existent falls short of meeting the requirements imposed on the underwater acoustic source of interest in this contract. However, modification of the device to more nearly fulfill these requirements seems entirely possible.

V. Appendix - Calculations on Hartmann Generator

A. Impedance at the Wall of a Radially Pulsating Cylinder in a Free Field Medium of Water

Consider the case shown in the sketch:



The general equation for the pressure in the free sound field surrounding the cylinder is

$$P = A\left[J_o\left(\frac{\omega r}{c}\right) + i\, N_o\left(\frac{\omega r}{c}\right)\right] e^{-i\omega t} \quad \text{-----} \quad (1)$$

P = instantaneous pressure at a point

ω = $2\pi f$; f = frequency of pulsations

r = radius to a point in the sound field

c = velocity of sound in water

A = complex amplitude constant

t = time

J_o and N_o are Bessel and Neumann Functions respectively

From momentum and continuity we know that

$$\rho \frac{\partial u}{\partial t} = -\frac{\partial p}{\partial r} \quad \text{----------------} \quad (2)$$

u = instantaneous velocity at a point in the field

ρ = density of fluid, water in this case

Combining equations (1) and (2) we find that

$$u = \frac{A}{\varrho c}\left[-N_1\left(\frac{\omega r}{c}\right) + i\, J_1\left(\frac{\omega r}{c}\right)\right] e^{-i\omega t} \quad ----(3)$$

J_1 and N_1 are Bessel and Neumann Functions respectively

Imposing the condition that when

$$r = a \quad ; \quad u = U_o\, e^{-i\omega t}$$

the quantity A can be evaluated and is found to be

$$A = \frac{\varrho c\, U_o}{\left[-N_1\left(\frac{\omega a}{c}\right) + i\, J_1\left(\frac{\omega a}{c}\right)\right]} \quad ------(4)$$

Hence

$$u = U_o \frac{\left[-N_1\left(\frac{\omega r}{c}\right) + i\, J_1\left(\frac{\omega r}{c}\right)\right]}{\left[-N_1\left(\frac{\omega a}{c}\right) + i\, J_1\left(\frac{\omega a}{c}\right)\right]} e^{-i\omega t} \quad ------(5)$$

and

$$p = \varrho c\, U_o \frac{\left[J_o\left(\frac{\omega r}{c}\right) + i\, N_o\left(\frac{\omega r}{c}\right)\right]}{\left[-N_1\left(\frac{\omega a}{c}\right) + i\, J_1\left(\frac{\omega a}{c}\right)\right]} e^{-i\omega t} \quad ------(6)$$

At the surface of the cylinder, $r = a$, so the reaction pressure there is given by

$$p = \varrho c\, U_o \frac{\left[J_o\left(\frac{\omega a}{c}\right) + i\, N_o\left(\frac{\omega a}{c}\right)\right]}{\left[-N_1\left(\frac{\omega a}{c}\right) + i\, J_1\left(\frac{\omega a}{c}\right)\right]} e^{-i\omega t} \quad ------(7)$$

This can be written as

$$p = \varrho c\, U_o\, e^{-i\omega t}\left[A - i B\right] \quad --------(8)$$

where

$$A = \frac{\left[N_o\left(\frac{\omega a}{c}\right) J_1\left(\frac{\omega a}{c}\right) - J_o\left(\frac{\omega a}{c}\right) N_1\left(\frac{\omega a}{c}\right)\right]}{\left[N_1\left(\frac{\omega a}{c}\right)\right]^2 + \left[J_1\left(\frac{\omega a}{c}\right)\right]^2} \quad ----(9)$$

And

$$B = \frac{\left[J_o\left(\frac{\omega a}{c}\right) J_1\left(\frac{\omega a}{c}\right) + N_o\left(\frac{\omega a}{c}\right) N_1\left(\frac{\omega a}{c}\right)\right]}{\left[N_1\left(\frac{\omega a}{c}\right)\right]^2 + \left[J_1\left(\frac{\omega a}{c}\right)\right]^2} \quad ---(10)$$

The specific acoustic impedance at the cylinder wall is given by

$$Z = \frac{\text{Pressure}}{\text{Velocity}}$$

$$Z = \rho c (A - iB) \qquad (11)$$

Z = specific acoustic impedance

Plots of A and B versus the quantity $(\frac{\omega a}{c})$ are shown in Figures 24 and 25.

The real part of Z, namely $A\rho c$ is the resistive component of the acoustic load apparent to the cylinder while the imaginary part of Z, namely $B\rho c$, is the reactive component.

At values of $(\frac{\omega a}{c})$ greater than 1 the impedance is essentially resistive, its value being very close to the characteristic acoustic impedance (ρc) of water. This resistive load is very high compared to the characteristic acoustic impedance of air. The condition that $\frac{\omega a}{c} \geq 1$
exists when $f \geq 9$ KCS.
for a 1 inch radius cylinder.

B. Impedance Apparent to a Cylindrical Wave Inside a Stiff Cylinder Containing Air

This analysis is intended to indicate the acoustic impedance apparent to a cylindrically radiating source with its center line positioned on the longitudinal center line of a steel cylinder surrounded by water. This system is representative of the Hartmann Generator in a cylindrical enclosure.

A rigid cylinder was assumed to represent the somewhat flexible steel cylinder surrounded by water, to simplify calculations. This assumption implies that the components of impedance at the inside wall of the cylinder are an infinite reactance and no

resistance when in reality they are a very large reactance and a very small resistance. The conditions which cause the impedance near the center of a rigid cylinder to be zero, closely approximate the conditions which cause a very low impedance near the center of the steel cylinder actually used. The purpose of this analysis is to evaluate these conditions.

The general expression for the pressure at a point in a cylindrical sound field made up of outgoing and incoming waves is

$$p = \left\{ D\left[J_0\left(\tfrac{\omega r}{c}\right) + i N_0\left(\tfrac{\omega r}{c}\right)\right] + E\left[J_0\left(\tfrac{\omega r}{c}\right) - i N_0\left(\tfrac{\omega r}{c}\right)\right] \right\} e^{-i\omega t} \quad --(12)$$

where D and E are complex amplitude constants

c = velocity of sound in medium, air in this case

other symbols used as before.

Combining equation (12) with equation (2), Page 38, we get

$$u = \frac{1}{\rho c} \left\{ D\left[N_1\left(\tfrac{\omega r}{c}\right) - i J_1\left(\tfrac{\omega r}{c}\right)\right] - E\left[N_1\left(\tfrac{\omega r}{c}\right) + i J_1\left(\tfrac{\omega r}{c}\right)\right] \right\} e^{-i\omega t} \quad --(13)$$

where ρ = density of air

If we now assume that the rigid cylinder wall is present at $r = a$ we can write that

$$u = 0 \qquad \text{when} \qquad r = a$$

Hence

$$E = D \, \frac{\left[N_1\left(\tfrac{\omega a}{c}\right) - i J_1\left(\tfrac{\omega a}{c}\right)\right]}{\left[N_1\left(\tfrac{\omega a}{c}\right) + i J_1\left(\tfrac{\omega a}{c}\right)\right]} \quad -----------(14)$$

It follows that

$$u = \frac{D e^{-i\omega t}}{\rho c} \left\{ N_1\left(\tfrac{\omega r}{c}\right)\left[1 - \frac{N_1\left(\tfrac{\omega a}{c}\right) - i J_1\left(\tfrac{\omega a}{c}\right)}{N_1\left(\tfrac{\omega a}{c}\right) + i J_1\left(\tfrac{\omega a}{c}\right)}\right] - i J_1\left(\tfrac{\omega r}{c}\right)\left[1 + \frac{N_1\left(\tfrac{\omega a}{c}\right) - i J_1\left(\tfrac{\omega a}{c}\right)}{N_1\left(\tfrac{\omega a}{c}\right) + i J_1\left(\tfrac{\omega a}{c}\right)}\right] \right\} -(15)$$

and

$$p = D e^{-i\omega t} \left\{ J_0\left(\tfrac{\omega r}{c}\right)\left[1 + \frac{N_1\left(\tfrac{\omega a}{c}\right) - i J_1\left(\tfrac{\omega a}{c}\right)}{N_1\left(\tfrac{\omega a}{c}\right) + i J_1\left(\tfrac{\omega a}{c}\right)}\right] + i N_0\left(\tfrac{\omega r}{c}\right)\left[1 - \frac{N_1\left(\tfrac{\omega a}{c}\right) - i J_1\left(\tfrac{\omega a}{c}\right)}{N_1\left(\tfrac{\omega a}{c}\right) + i J_1\left(\tfrac{\omega a}{c}\right)}\right] \right\} \quad --(16)$$

Hence, the specific acoustic impedance inside a stiff cylinder of radius a at some radius r from the center line is given by

$$z = \frac{p}{u} = \rho c \left\{ \frac{\left[J_0\left(\frac{\omega r}{c}\right)N_1'\left(\frac{\omega a}{c}\right) - N_0\left(\frac{\omega r}{c}\right)J_1'\left(\frac{\omega a}{c}\right)\right] + i\left[N_0\left(\frac{\omega r}{c}\right)J_1'\left(\frac{\omega a}{c}\right) - J_0\left(\frac{\omega r}{c}\right)N_1'\left(\frac{\omega a}{c}\right)\right]}{\left[N_1\left(\frac{\omega r}{c}\right)J_1'\left(\frac{\omega a}{c}\right) - J_1\left(\frac{\omega r}{c}\right)N_1'\left(\frac{\omega a}{c}\right)\right] - i\left[J_1\left(\frac{\omega r}{c}\right)N_1'\left(\frac{\omega a}{c}\right) - N_1\left(\frac{\omega r}{c}\right)J_1'\left(\frac{\omega a}{c}\right)\right]} \right\} \quad (17)$$

The value of a for our design was 1 inch while the value of r (not accurately known) was estimated to be about .05 inches (the radius of the cavity resonator). With these values established the impedance apparent to the generator (at a radius of .05 inches) was known as a function of frequency. A plot of this impedance versus frequency is shown in Figure 26.

At frequencies of 9.2, 16 and 22.8 KCS the impedance apparent to the Hartmann Generator as predicted by this analysis is zero. In reality this impedance is very low (but not zero) at these points, the condition which should favor good operation of the generator.

C. Sample Calculation of Power Output from Pressure Amplitude Measurement

Since pressure amplitude at a known point in the sound field is the quantity commonly measured, the power being radiated by an acoustic source must be calculated from this measurement. Our estimates of radiated power are primarily intended to indicate order of magnitude, so the following simplifying assumptions have been made:

1. The transducer radiates to a free field.
2. A simple sound field has been assumed for each device, a

cylindrical pattern for the Hartmann Generator, a hemispherical pattern for the water hammer device.

3. The pressure variation is sinusoidal.

4. Measurements of pressure amplitude were taken at points far enough away from the source so that pressure and velocity were in phase and the wave front was approximately a plane.

With these assumptions the relationship between intensity and pressure is

$$\Upsilon = \frac{(P_{RMS})^2}{\rho C} = \frac{(P_{MAX})^2}{2\rho C} \qquad \text{------ (18)}$$

Υ = intensity

P_{RMS} = root mean square pressure amplitude

P_{MAX} = peak pressure amplitude

ρ = density of fluid medium

C = velocity of sound in fluid medium

Power and intensity are simply related by

$$\Pi = \Upsilon A \qquad \text{---------- (19)}$$

where

$A = 2\pi r^2$ for a hemispherical radiation pattern

or

$A = 2\pi r h$ for a cylindrical radiation pattern

Π = radiated power

A = area over which intensity was distributed

r = distance from source to point where pressure was measured

h = height of cylindrical radiation pattern

Combining equations 18 and 19 for the case of a cylindrical pattern

$$\Pi = \frac{(P_{RMS})^2}{\rho c}(2\pi r h) = \frac{(P_{MAX})^2}{\rho c}(\pi r h) \text{ -----(20)}$$

Considering the values obtained from the measurement (reported on page 10 of the report) of the Hartmann Generator No. 1 output in free sound field of air:

$$P_{RMS} = 6{,}000 \ \frac{\text{dyne}}{\text{cm}^2}$$

r = 4.5 inches = 11.5 cms.

h = 3 inches = 7.6 cms.

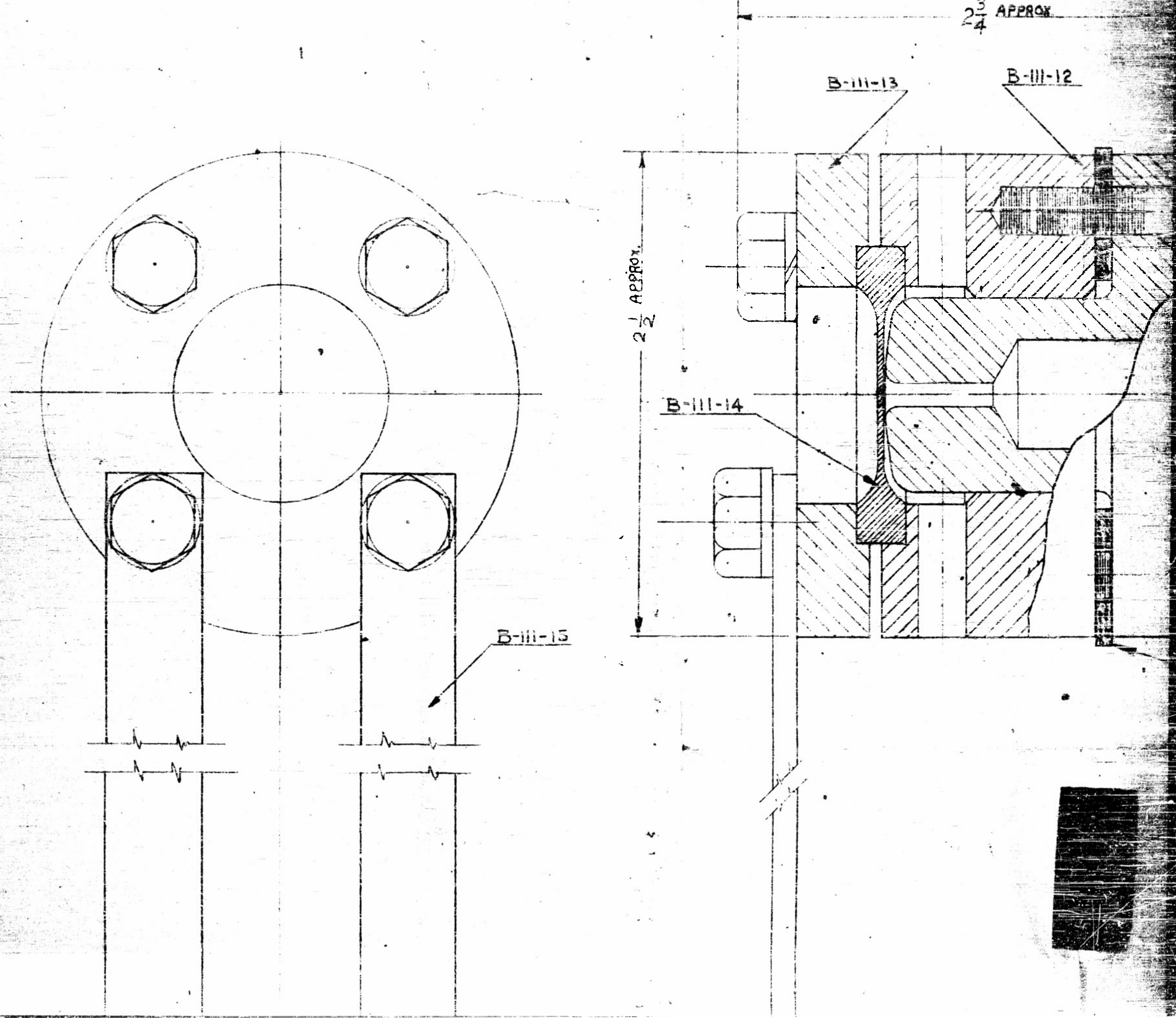
$$(\rho c) \text{ air} = 42 \ \frac{\text{gms}}{\text{sec.cm}^2}$$

$$\Pi = \frac{(6000)^2}{42}(2\pi)(11.5)(7.6)$$

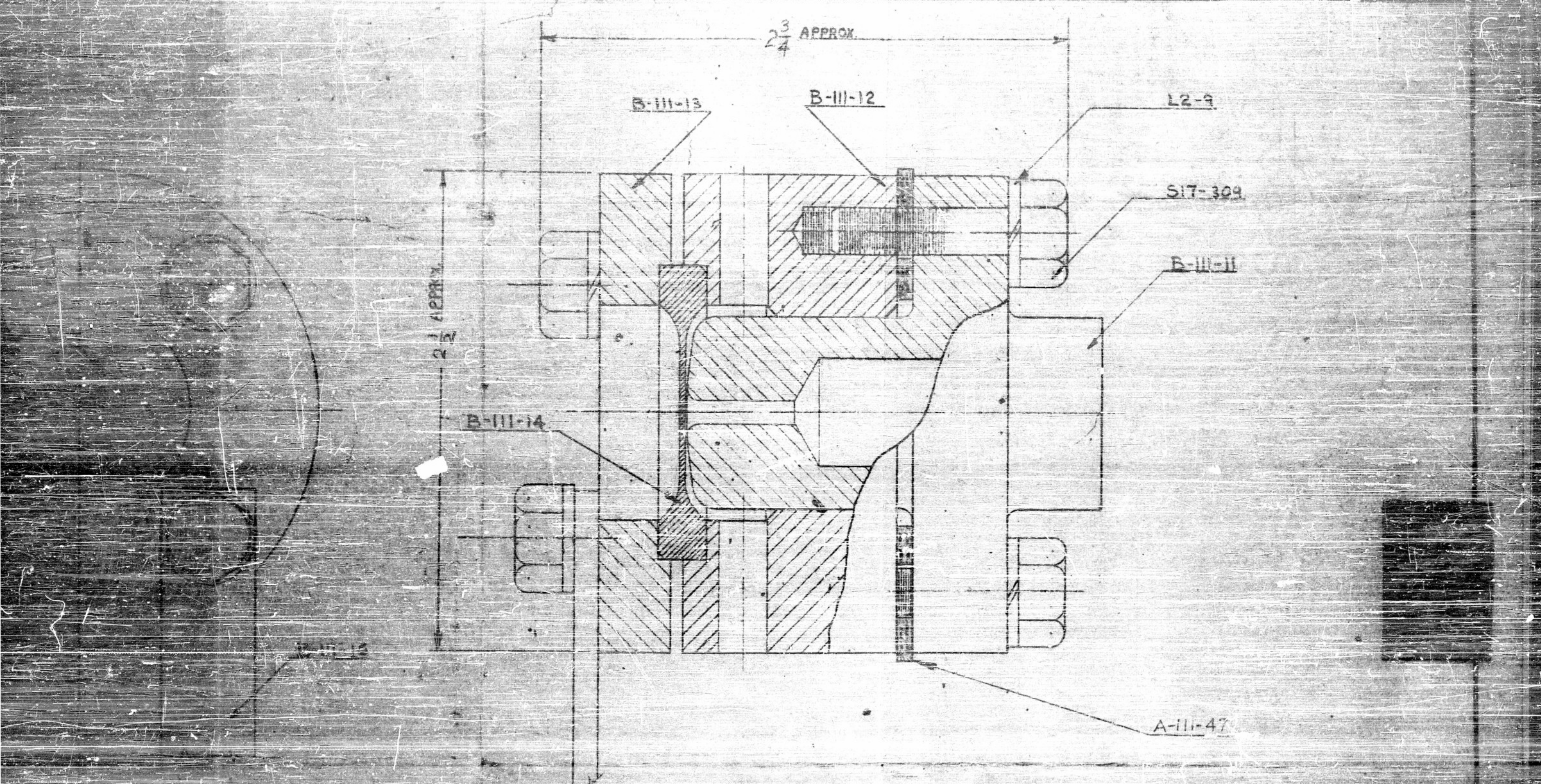
$$\Pi = 47 \times 10^7 \ \frac{\text{ergs}}{\text{sec}}$$

$$\Pi = 47 \text{ watts}$$

C-111-10



$2\frac{3}{4}$ APPROX
B-111-13
B-111-12
$2\frac{1}{2}$ APPROX
B-111-14
B-111-15



PART NO.	DESCRIPTION	MAT'L	QTY
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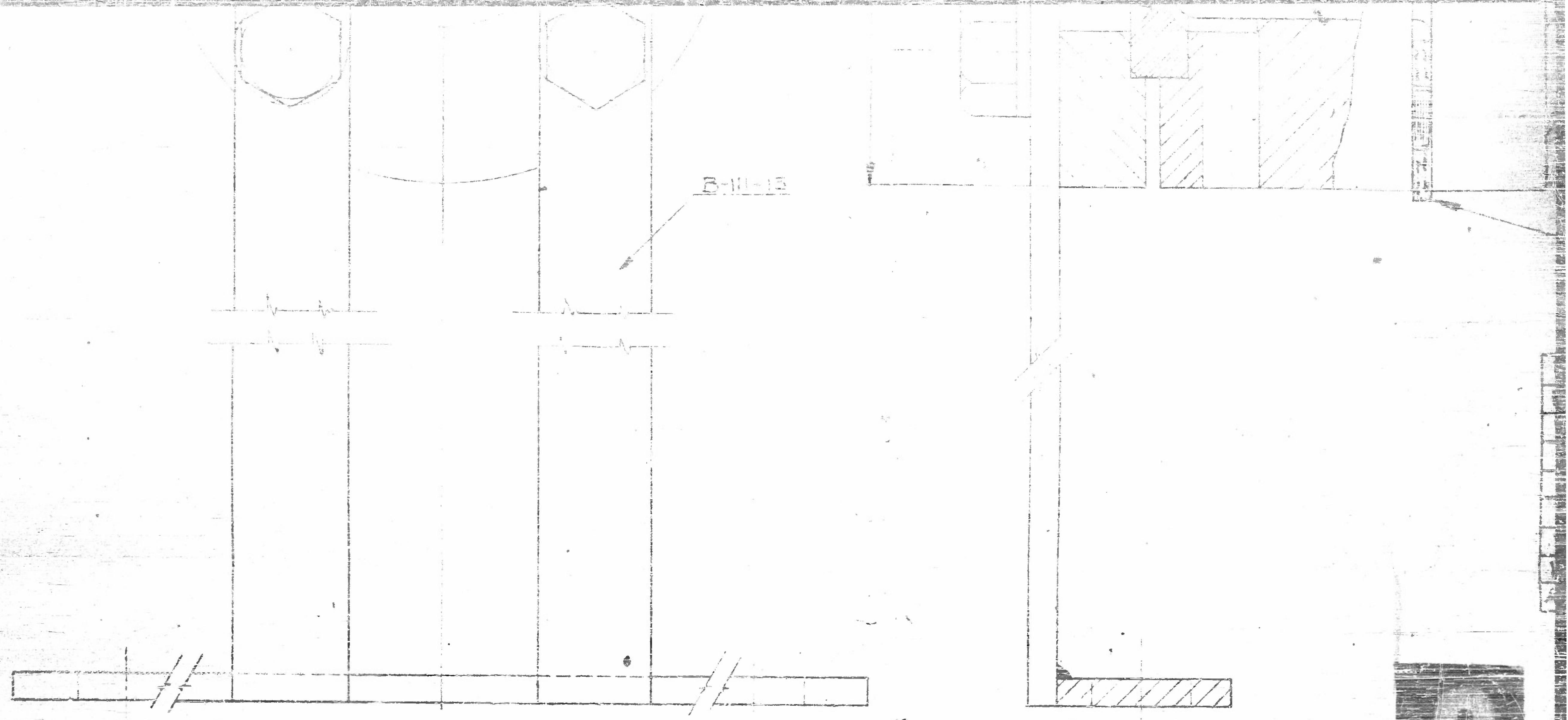
B-111-12	NOZZLE BLOCK	ST.ST.	1
B-111-13	CLAMPING RING	ST.ST.	1
B-111-14	DIAPHRAGM	[illegible]	1
B-111-15	BRACKET	[illegible]	1
S17-309	1/4 X1 LONG HEX. HD. MACH. SC.	ST.ST.	2
L2-9	1/4" LOCK WASHER	ST.ST.	[illegible]
A-111-47	RUBBER SPACER	[illegible]	[illegible]



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REVISIONS

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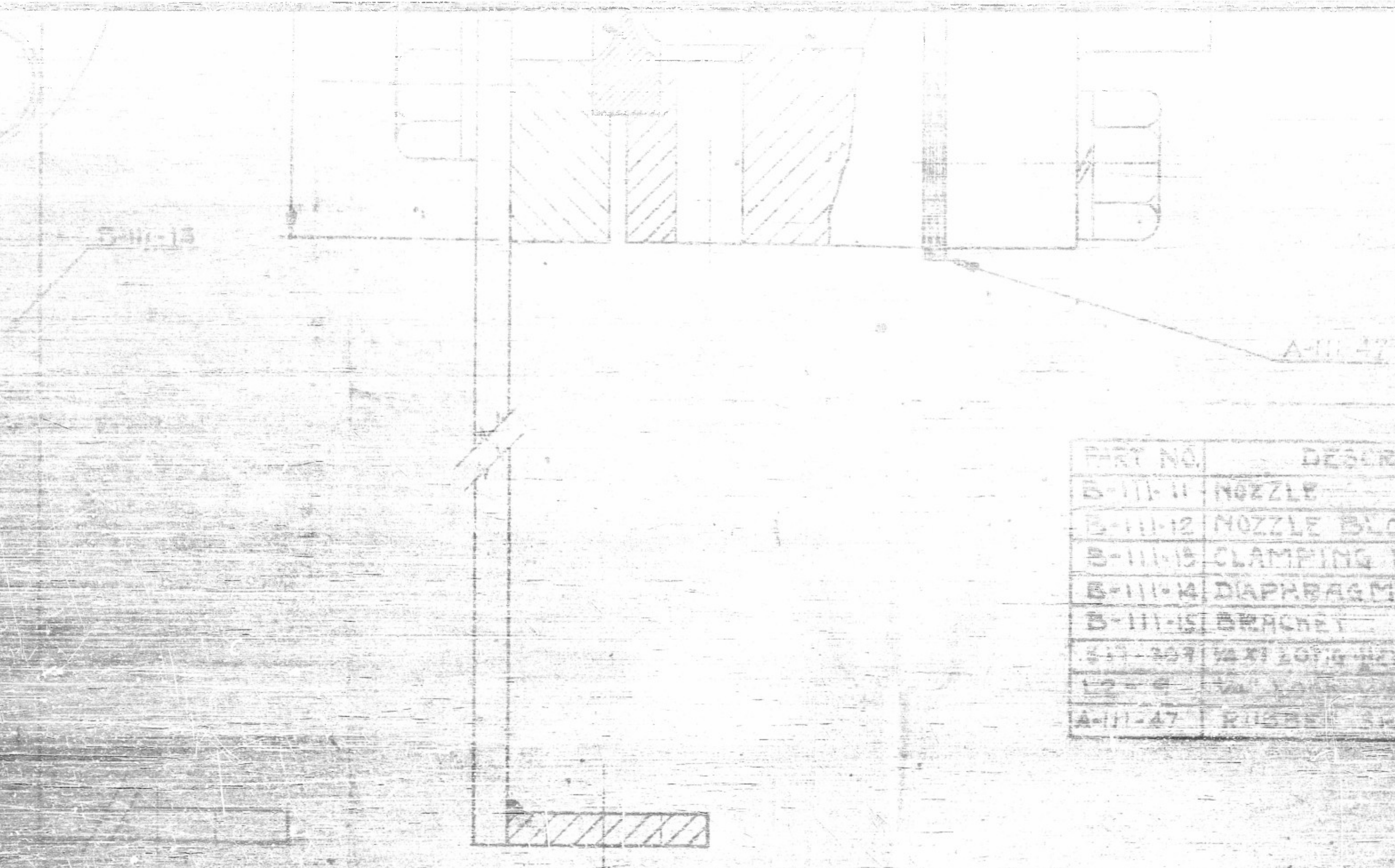


PART NO.	DESCRIPTION
B-111-11	NOZZLE
B-111-12	NOZZLE BLOCK
B-111-13	CLAMPING RING
B-111-14	DIAPHRAGM
B-111-15	BRACKET
[illegible]	[illegible]
[illegible]	[illegible]
A-111-47	RUBBER [illegible]

REVISIONS

NO.	ITEM	DATE	NO.	ITEM	DATE
3	[illegible] 873	[illegible]			

DR. E.H.L. DATE 2-1-52

SCALE: TWICE

MATERIAL: —

FOR ONR-111

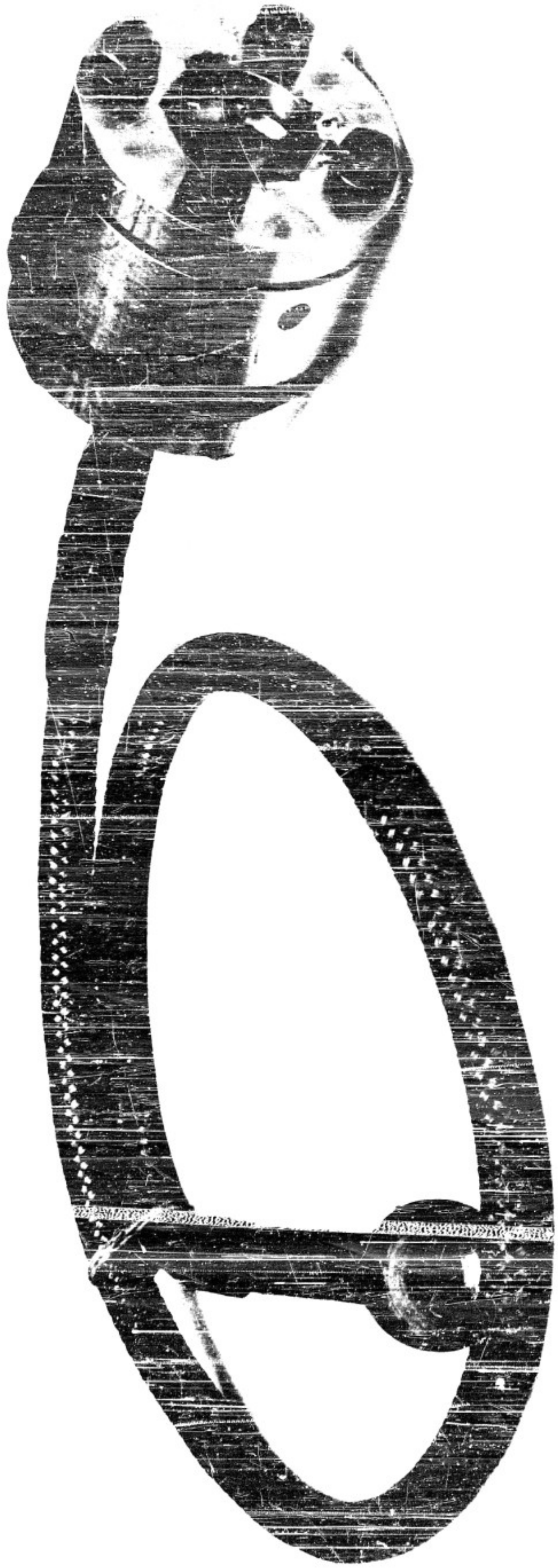
NAME: SUB. ASSY.-AIR-WATER [illegible]

ULTRASONIC CORPORATION

CAMBRIDGE, MASS.

FLUID DRIVEN DIAPHRAGM ASSEMBLED FOR TESTING

FIGURE 2



PARTS OF FLUID DRIVEN DIAPHRAGM

NOZZLE | NOZZLE BLOCK | DIAPHRAGM | CLAMPING RING

FIGURE 3

ULTRASONIC CORPORATION CAMBRIDGE, MASS.	TITLE: OIL FLOW SYSTEM FOR FLUID DRIVEN DIAPHRAGM	BY:	DATE:

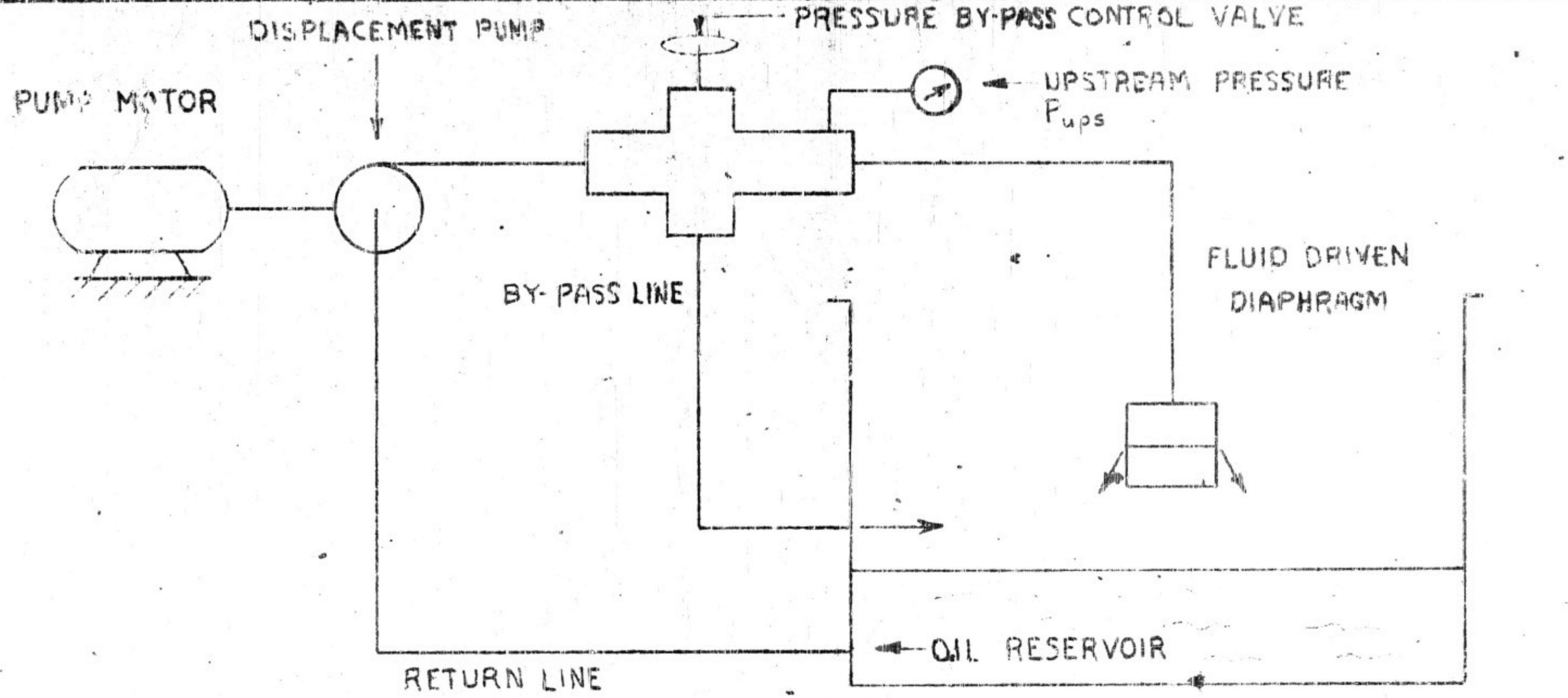


FIGURE 4

ULTRASONIC CORPORATION CAMBRIDGE, MASS.	TITLE: WATER FLOW SYSTEM FOR FLUID DRIVEN DIAPHRAGM	BY: A.M.C.	DATE: 8/28/52

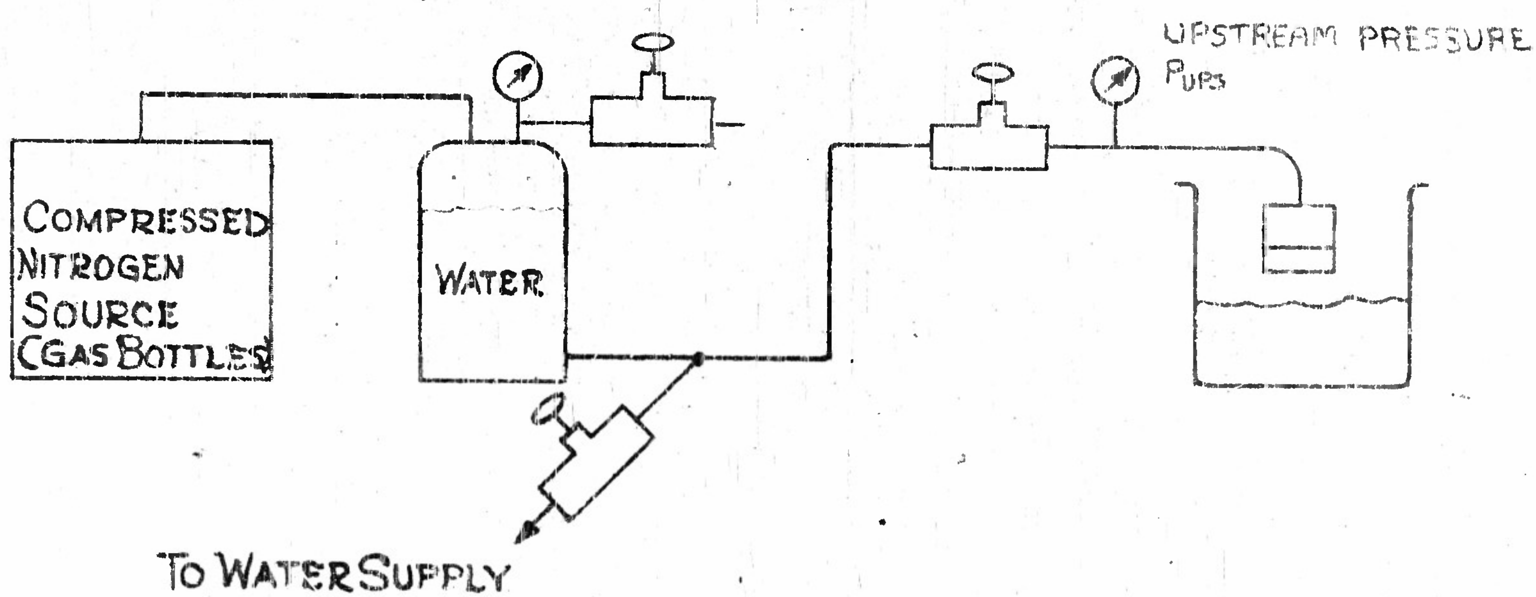
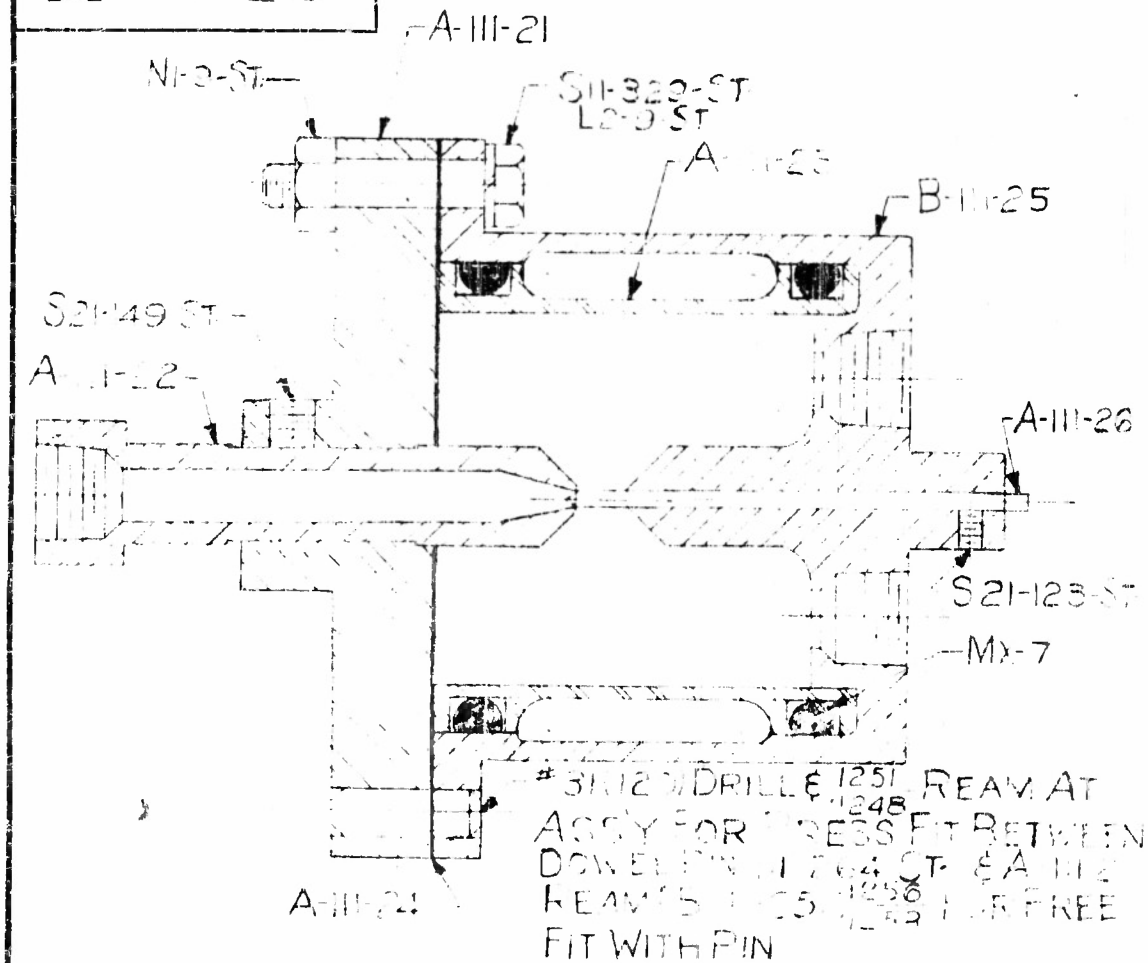


FIGURE-5

PRINTED BY SPAULDING MOSS CO., BOSTON, MASS., RE-ORDER NO. A-1361

A-111-20



PART NO.	DESCRIPTION	MAT'L	QUAN
B-111-25	HOUSING	BRASS	1
A-111-21	CAP HOUSING	BRASS	1
A-111-22	NOZZLE	BRASS	1
A-111-23	CYLINDRICAL RESONATOR	ST. ST	1
A-111-24	GASKET [illegible]	[illegible]	1
S21-123	[illegible] LG SCR, SOC SET	STEEL	[illegible]
S21-149	1/4-20NC [illegible]	STEEL	[illegible]
S[illegible]29	[illegible]	STEEL	[illegible]
L2-0	[illegible]	STEEL	4
NI-2	[illegible] HEX [illegible]	STEEL	4
MX-7	#33 [illegible] PACKING [illegible] / #[illegible]OLINE AIR [illegible]	RUBBER	2
DI-[illegible]-CT	[illegible] PIN [illegible]	STEEL	2
A-111-26	PISTON [illegible]	CRS	1

ALIGN [illegible] B-111-25 & A-111-22 FOR MAXIMUM ECCENTRICITY OF .004 BEFORE DOWELING

[illegible]

ALL DIMENSIONS ARE IN INCHES UNLESS OTHERWISE SPECIFIED
TOLERANCE ON FRACTIONAL DIMENSIONS ± [illegible]
BREAK SHARP CORNERS UNLESS OTHERWISE SPECIFIED

REVISIONS

NO	WAS	DATE	NO	WAS	DATE
[illegible]	[illegible]				

REDRAWN | DATE | WEIGHT

DR. [illegible] DATE [illegible] CKD. [illegible] DATE 2-26-52 APPD DATE

SCALE: FULL

MATERIAL:

FINISH:

FOR: CONTRACT # 111

NAME: SUB ASS'Y HARTMANN GENERATOR #1

ULTRASONIC CORPORATION
CAMBRIDGE, MASS.

A-111-20

ULTRASONIC CORPORATION CAMBRIDGE, MASS.	TITLE: SCHEMATIC OF HARTMANN GENERATOR #1	BY:	DATE:

PRINTED BY SPAULDING-MOSS CO., BOSTON, MASS. RE ORDER NO. A 1348

ULTRASONIC CORPORATION CAMBRIDGE, MASS.	TITLE: FREQUENCY VERSUS CAVITY LENGTH FOR HARTMANN GENERATOR #1	BY: AMC	DATE: 8/22/52

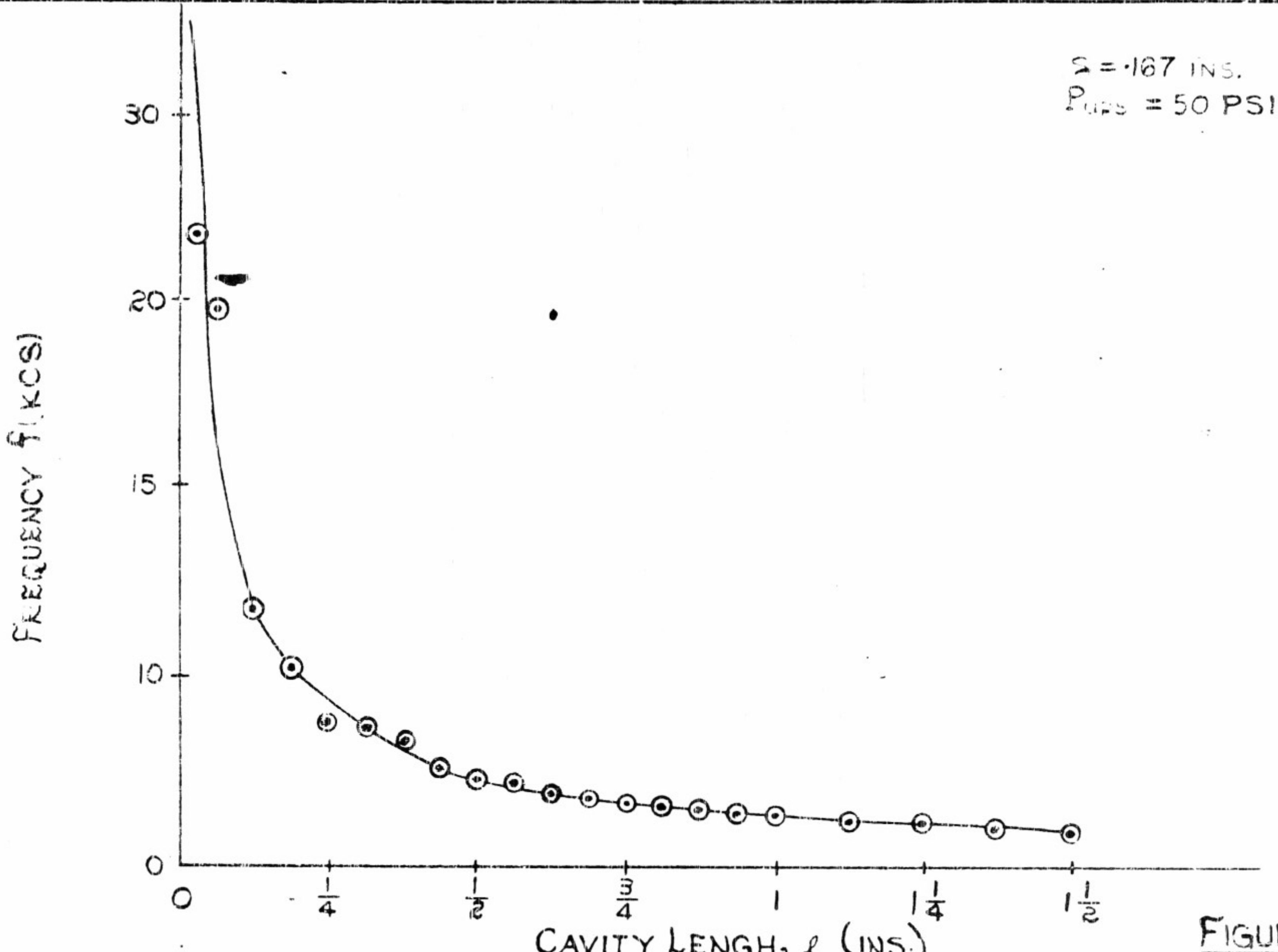


FIGURE-8

PRINTED BY SPAULDING-MOSS CO., BOSTON, MASS., RE-ORDER NO. A-1348

ULTRASONIC CORPORATION CAMBRIDGE, MASS.	TITLE: RMS PRESSURE AMPLITUDE VERSUS CAVITY LENGTH FOR HARTMANN GENERATOR #1	BY: Amc.	DATE: 8/22/52

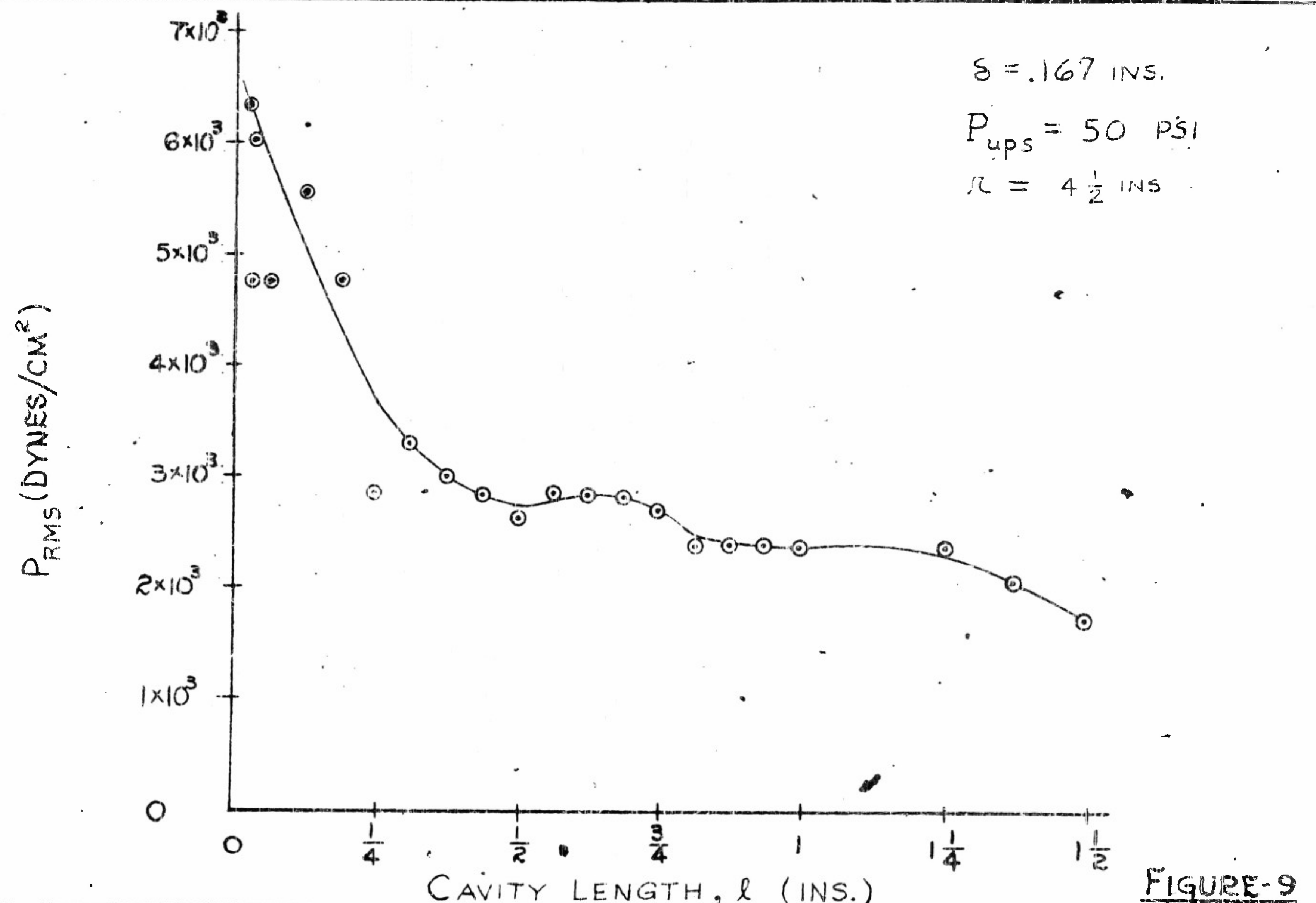


FIGURE-9

PRINTED BY SPAULDING-MOSS CO., BOSTON, MASS. RE-ORDER NO A-1340

B-111-30

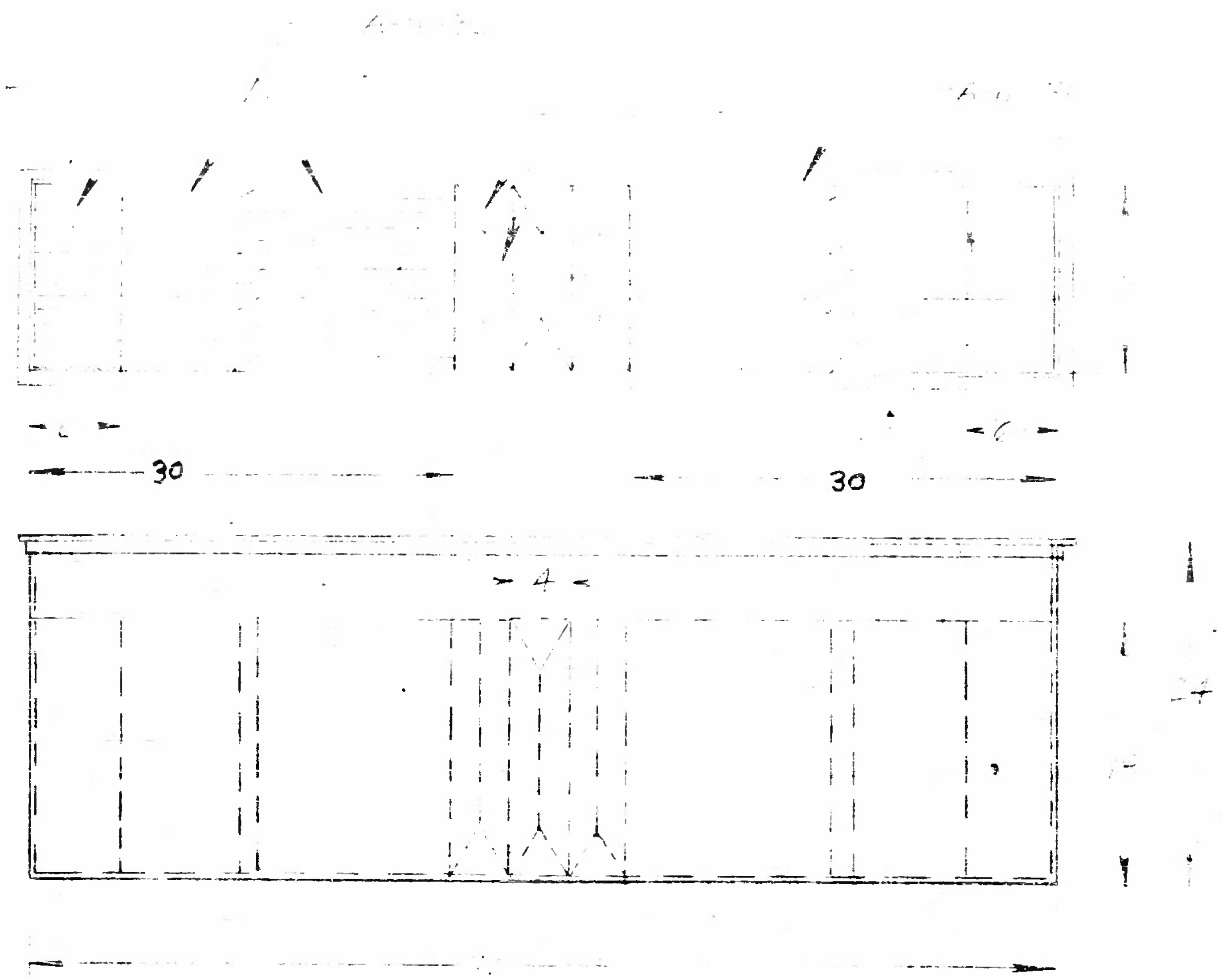


FIGURE 10

ALL DIMENSIONS ARE IN INCHES UNLESS OTHERWISE SPECIFIED
TOLERANCE ON FRACTIONAL DIMENSIONS ± 1/64
BREAK SHARP CORNERS UNLESS OTHERWISE SPECIFIED

REVISIONS

NO	WAS	DATE	NO	WAS	DATE

REDRAWN DATE WEIGHT

DR. [illegible] DATE [illegible] CKD. DATE APPD. DATE

SCALE: 1" = 1'

MATERIAL:

FINISH:

FOR: ONR

NAME: ANECHOIC TANK FOR UNDERWATER TESTING

ULTRASONIC CORPORATION
CAMBRIDGE, MASS.

B-111-30

PRINTED BY SPAULDING-MOSS CO. BOSTON, MASS.

ANECHOIC TANK ASSEMBLED FOR TESTING

FIGURE 11

ULTRASONIC CORPORATION CAMBRIDGE, MASS.	TITLE: MODIFIED HARTMANN GENERATOR #1	BY: C / W	DATE: 6-24-52

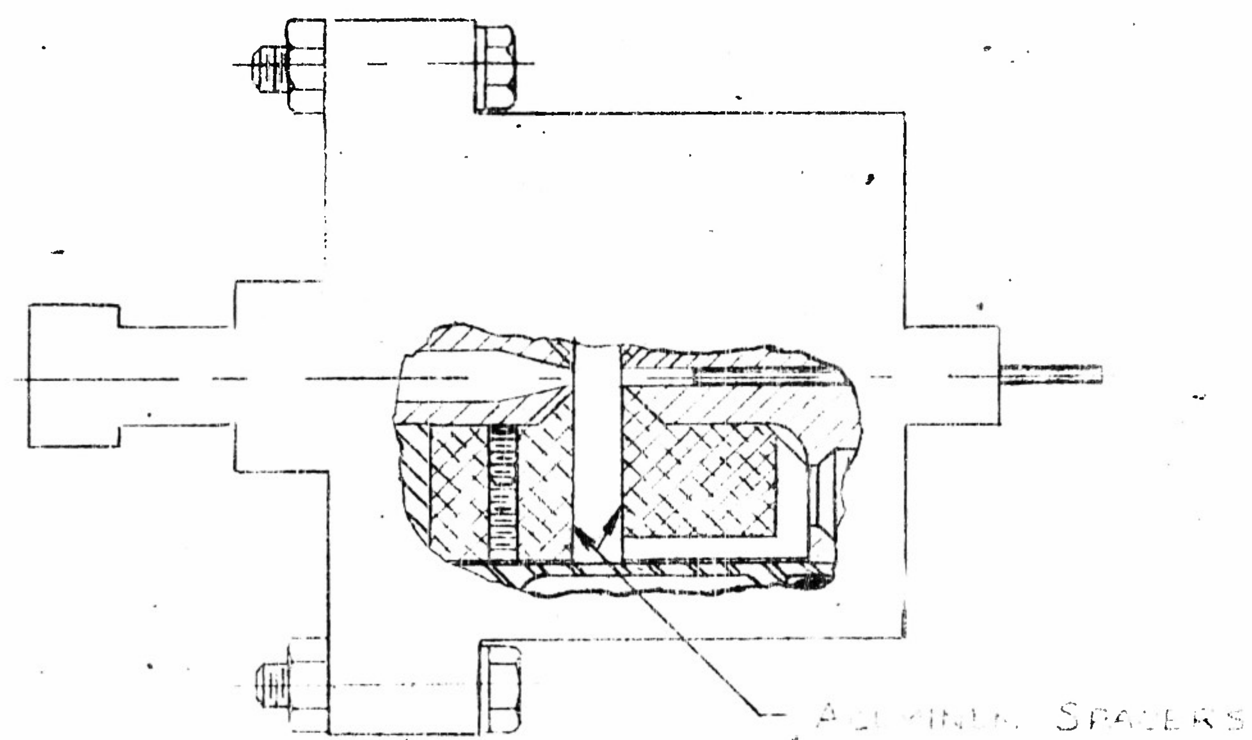
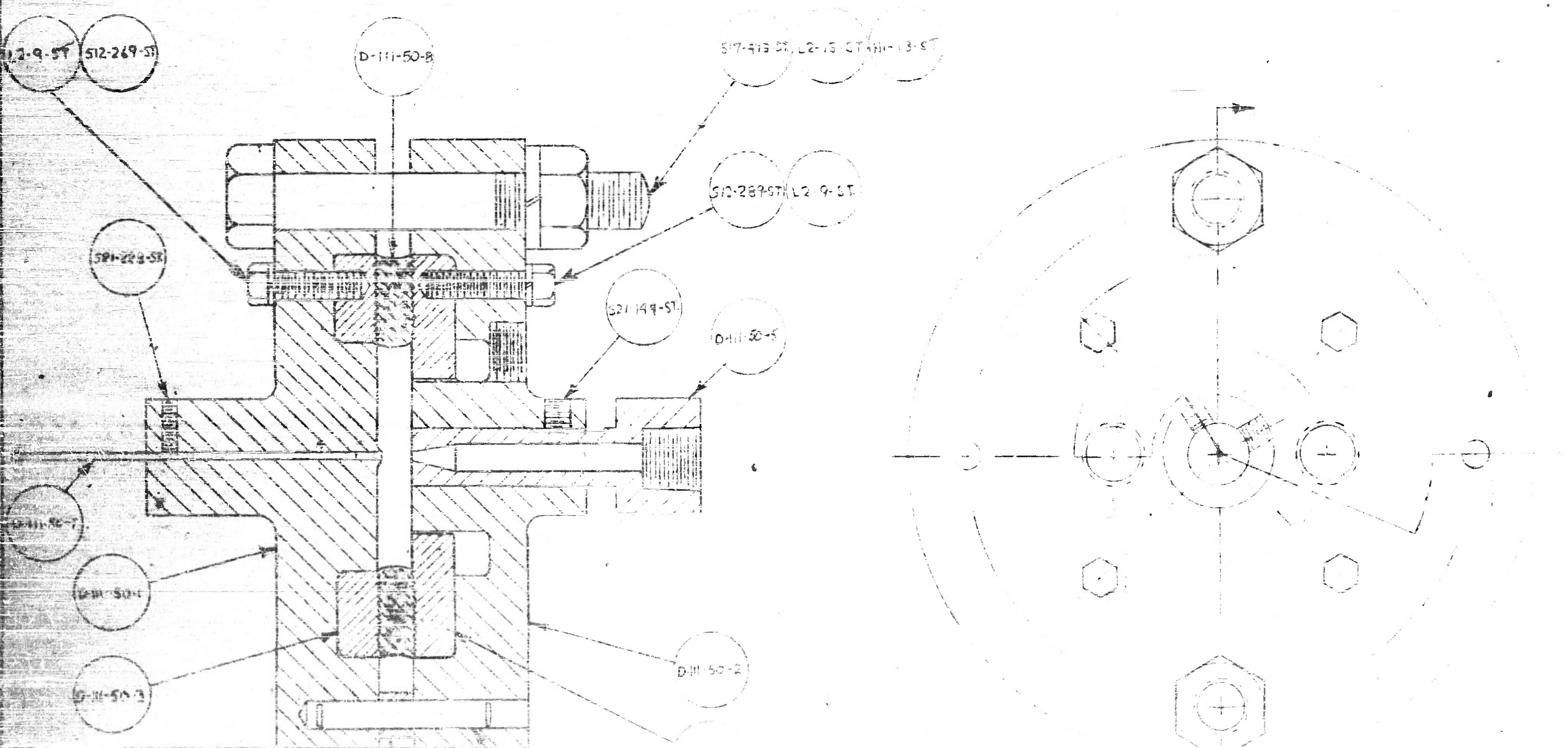


FIGURE 12

PRINTED BY SPAULDING-MOSS CO., BOSTON, MASS. RE-ORDER NO. A-1348



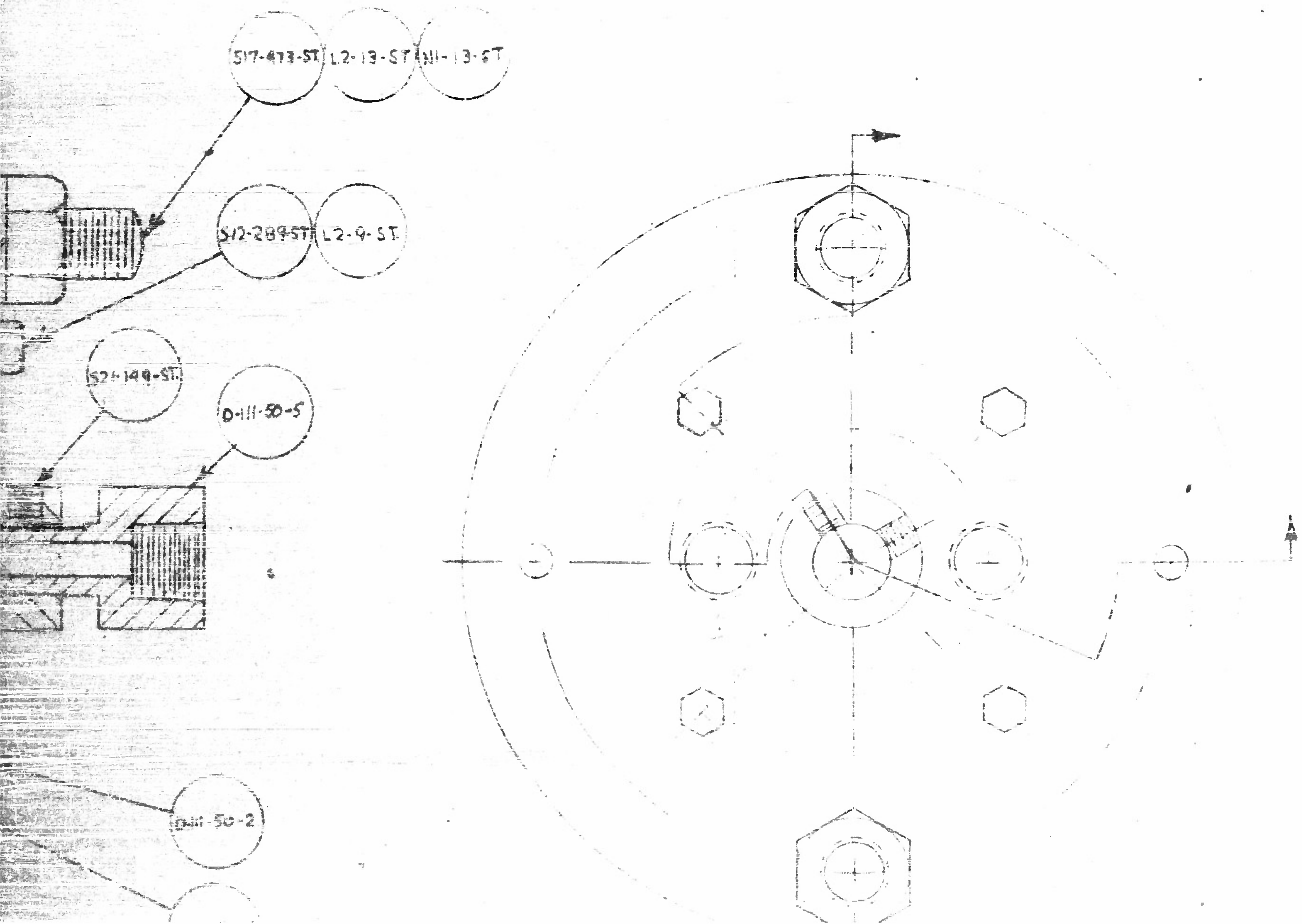
PART NO
D-111-50-
D-111-50-
D-111-50-
D-111-50-
D-111-50-
D-111-50-
D-111-50-
S12-289-
L2-9-ST
S12-269-
S17-473-
L2-13-S
N1-13-S
S21-149-
S21-223-
D1-349-
D-111-50

REVISIONS

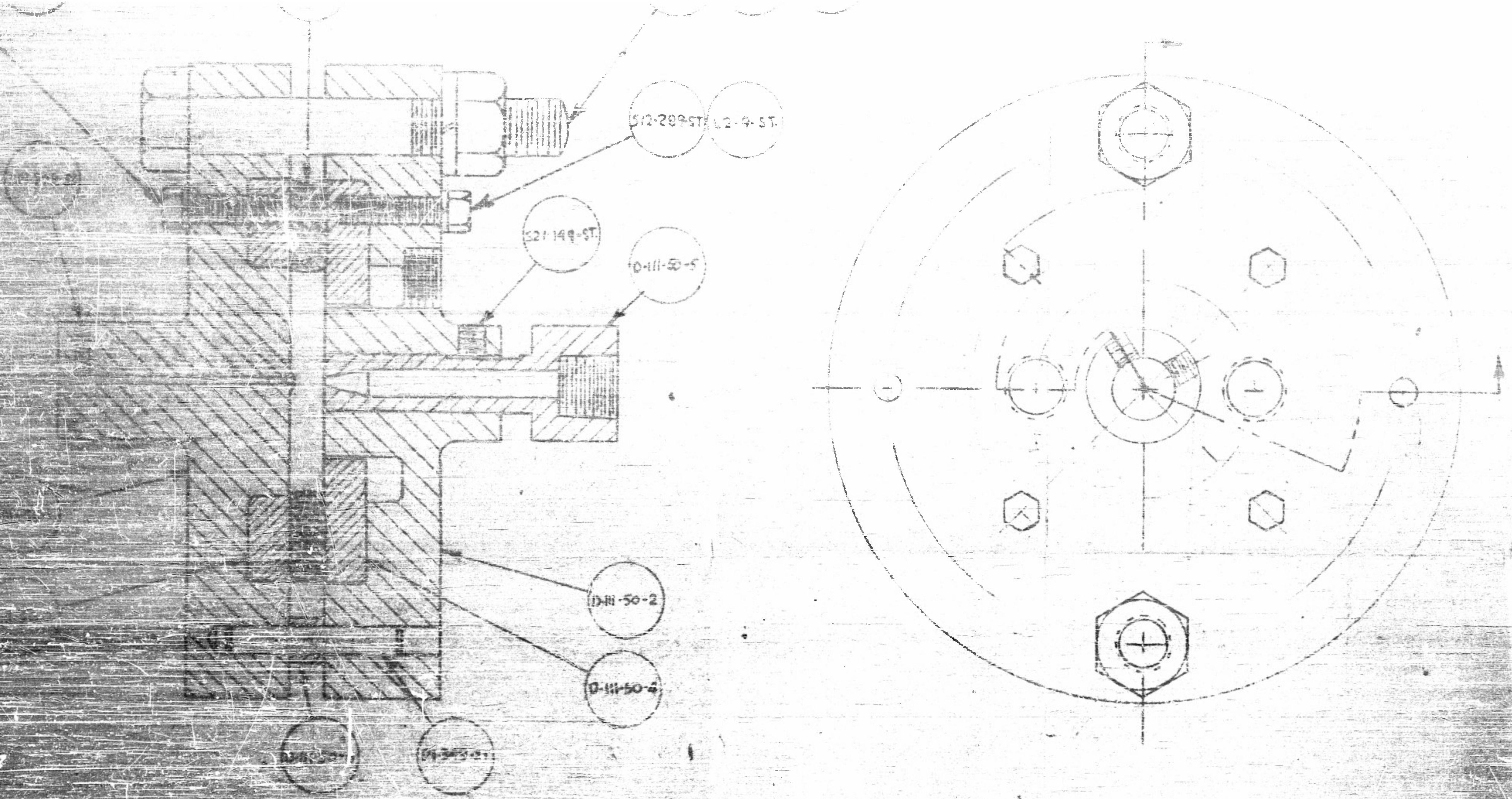
SYM.	DESCRIPTION	DATE	APPROVAL

PARTS LIST

PART NO.	DESCRIPTION	NO. REQ	MAT'L
D-111-50-1	CAVITY HOUSING	1	BR.
D-111-50-2	NOZZLE HOUSING	1	BR.
D-111-50-3	SEALING RING	1	BR.
D-111-50-4	SEALING RING	1	BR.
D-111-50-5	NOZZLE	1	BR.
D-111-50-6	SPACER	2	BR.
D-111-50-7	[illegible] ADJ. ROD $\frac{3}{16}$ DRILL ROD X [illegible] LG	1	ST.
S12-289-ST	SCREW - $\frac{1}{4}$ - 28 X $\frac{5}{8}$ LG HEX HD CAP	4	ST.
L2-9-ST	LOCKWASHER - $\frac{1}{4}$ STD.	8	ST
S12-291-ST	SCREW - $\frac{1}{4}$ - 28 X $\frac{3}{4}$ LG HEX HD CAP	4	ST.
S17-473-ST	BOLT - $\frac{1}{2}$ - 13 X 3" LG HEX HD MACH.	2	ST.
L2-13-ST	LOCKWASHER - $\frac{1}{2}$ STD.	2	ST
N1-13-ST	NUT - $\frac{1}{2}$ - 13 HEX	2	ST.
S21-149-ST	SET SCREW - $\frac{1}{4}$ - 20 X $\frac{1}{4}$ LG SOC. HD	2	ST
S21-223-ST	SET SCREW - # 4 - 40 X $\frac{1}{8}$ LG	1	ST.
D1-349-ST.	DOWEL PIN - $\frac{1}{4}$ D X $1\frac{1}{2}$ LG	2	ST
D-111-50-8	2" D X $3\frac{1}{2}$" D X 0.300 THK MOULDED RUBBER CYLINDER	1	RBR



2



ALL DIMENSIONS ARE IN INCHES [illegible]

[illegible]

REVELATION OF THE CONTENTS OF THIS [illegible] TO ANY UNAUTHORIZED PERSON IS [illegible] BY LAW.

[illegible]

DO NOT SCALE [illegible]

[illegible]	[illegible]	[illegible]	ST
[illegible]	[illegible]	1	ST
[illegible]	[illegible]	2	ST
[illegible]	[illegible]	1	RBR

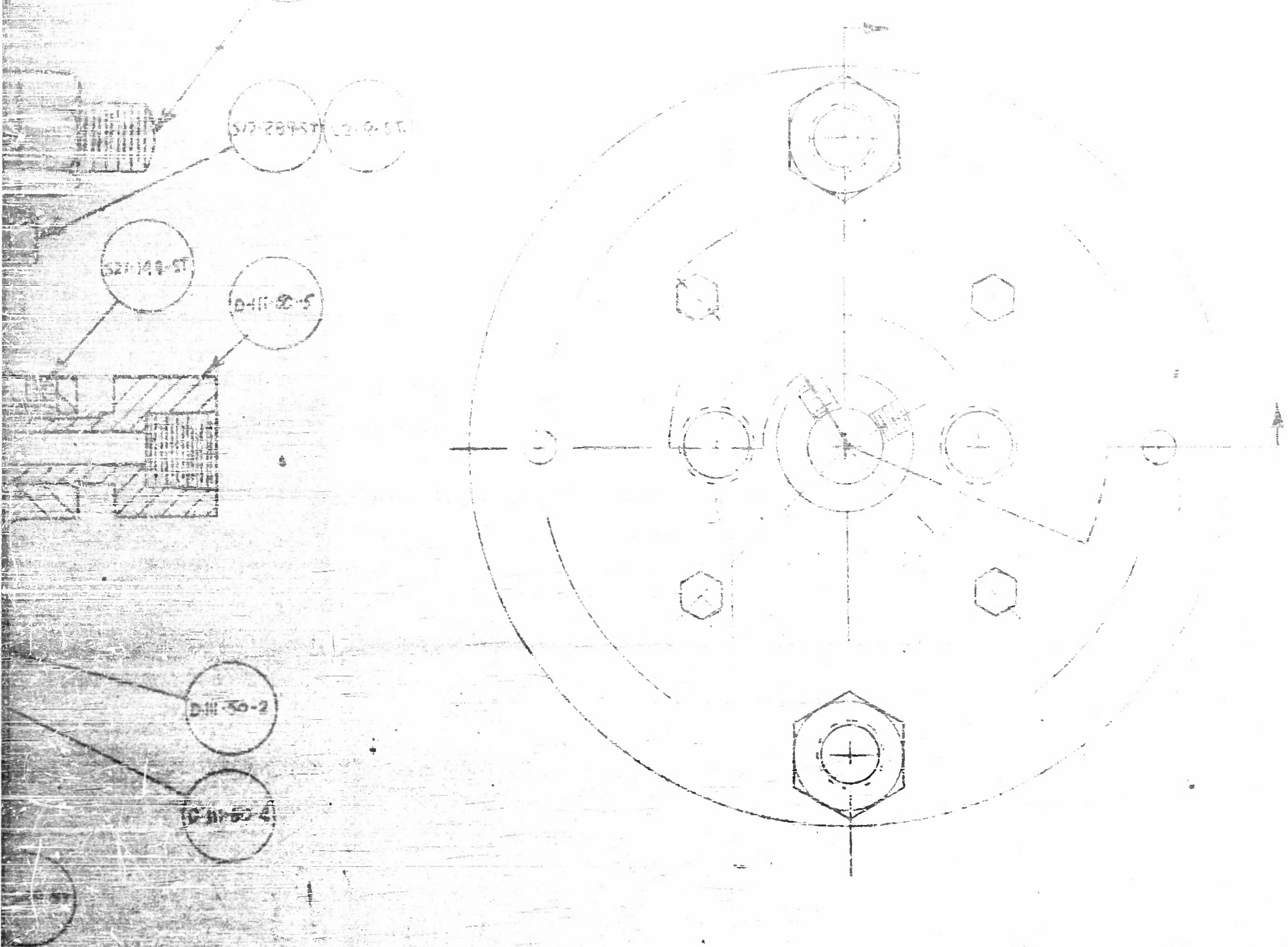
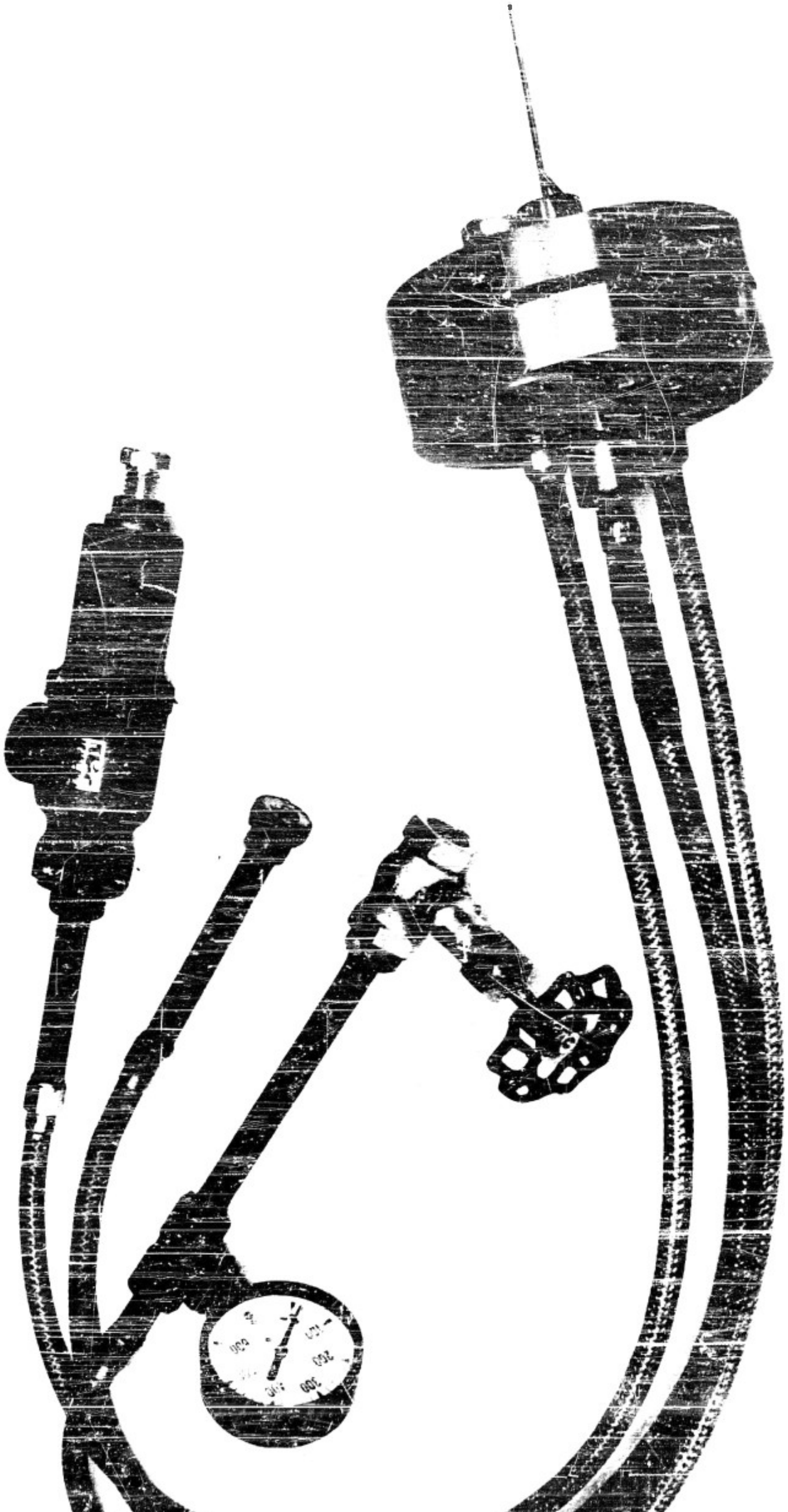


FIGURE 13

ALL DIMENSIONS ARE IN INCHES UNLESS OTHERWISE SPECIFIED TOLERANCE ON FRACTIONAL DIMENSIONS ±1/64 TOLERANCE ON DECIMAL DIMENSIONS ±.005 TOLERANCE ON ANGULAR DIMENSIONS [illegible] BREAK SHARP CORNERS UNLESS OTHERWISE SPECIFIED	WEIGHT	DR. [illegible] DATE [illegible]	CKD. DATE	APPD. DATE	
	SCALE: FULL	MATERIAL: ~		FINISH: ~	
USED ON	FOR: CONTRACT # 111				
	NAME: ASSY HARTMANN GENERATOR # 2				
DO NOT SCALE THIS DRAWING	ULTRASONIC CORPORATION CAMBRIDGE, MASS.				

HARTMANN GENERATOR NO. 2 ASSEMBLED FOR TESTING

FIGURE 14



PARTS OF HARTMANN GENERATOR NO. 2

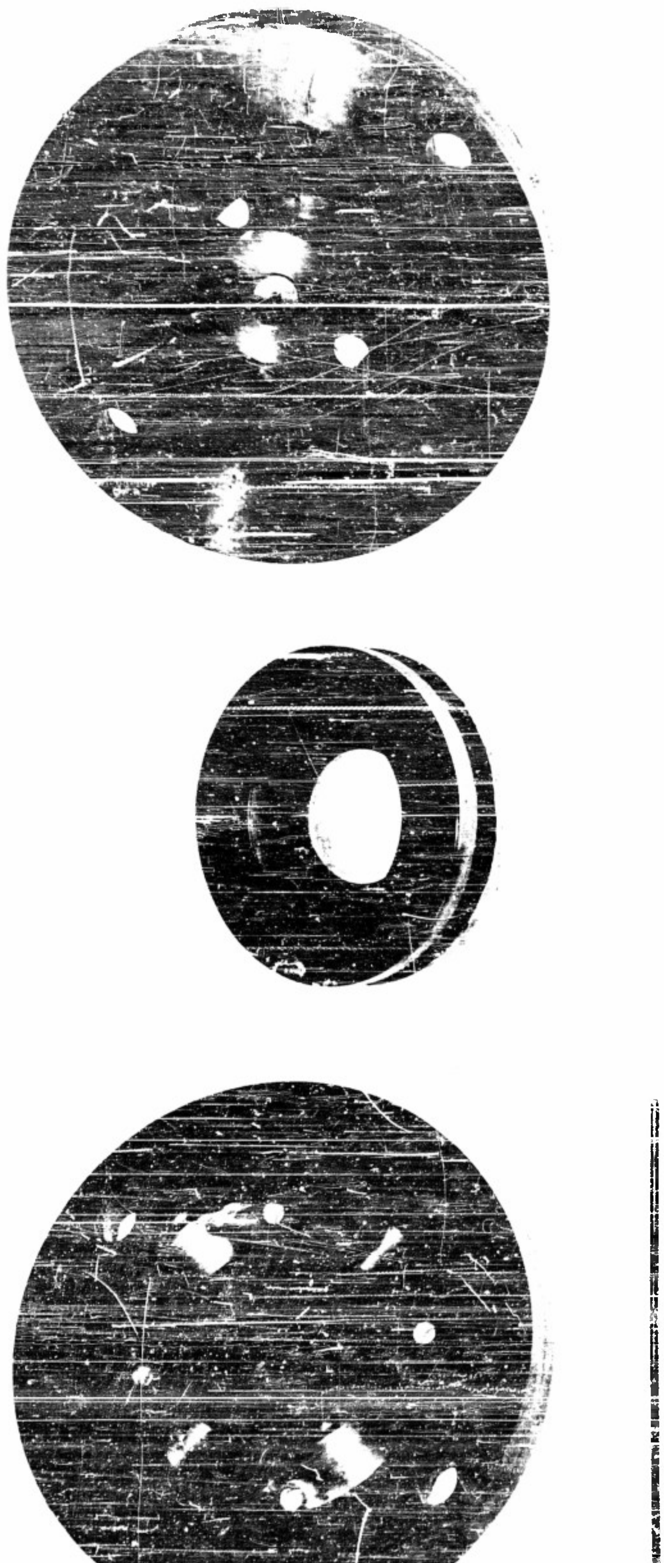
CAVITY HOUSING

RUBBER CYLINDER UNIT

NOZZLE HOUSING AND NOZZLE

CAVITY ADJUSTMENT ROD

FIGURE 15



ULTRASONIC CORPORATION CAMBRIDGE, MASS.	TITLE: SCHEMATIC OF HARTMANN GENERATOR # 2	BY:	DATE:

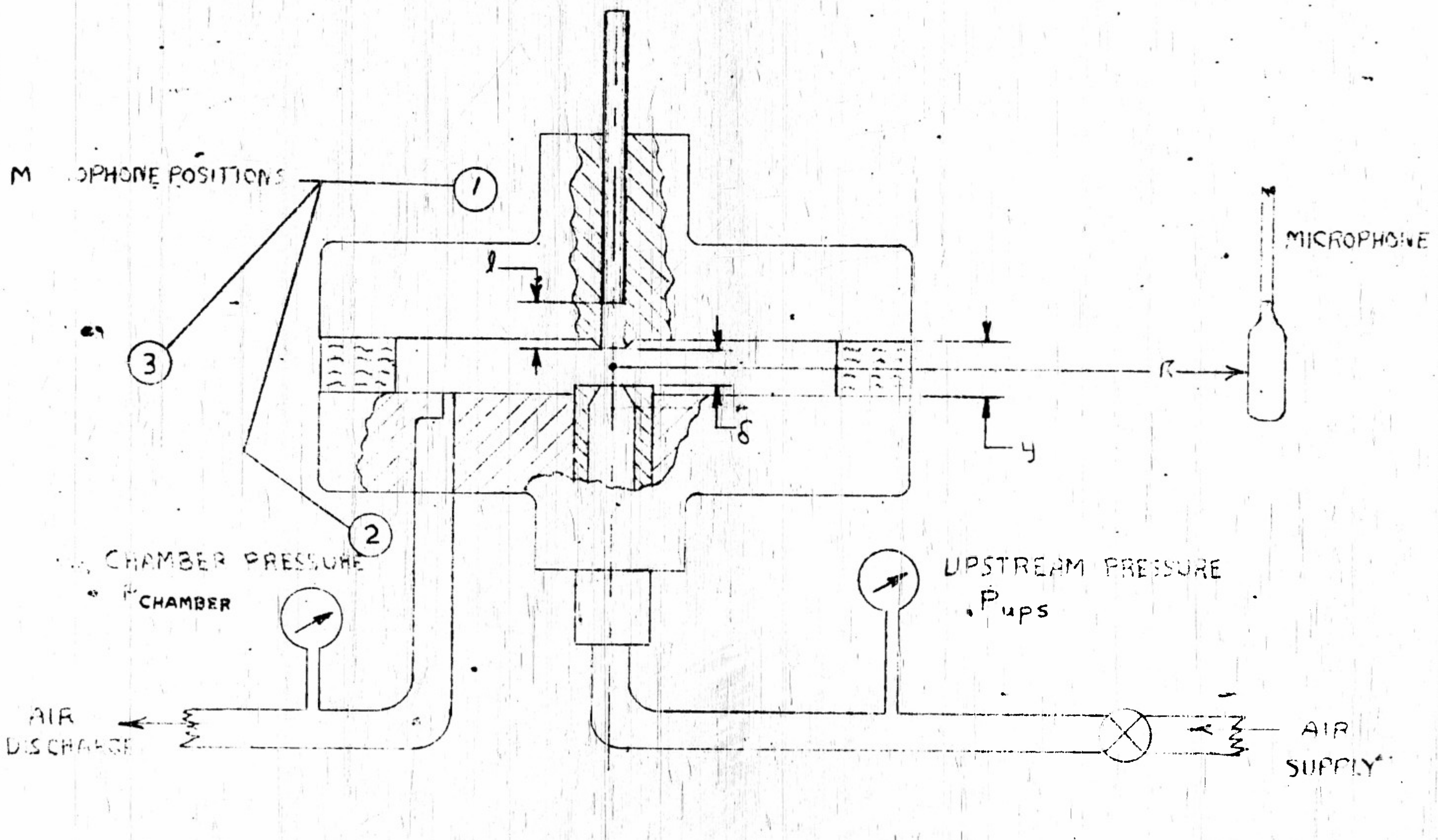
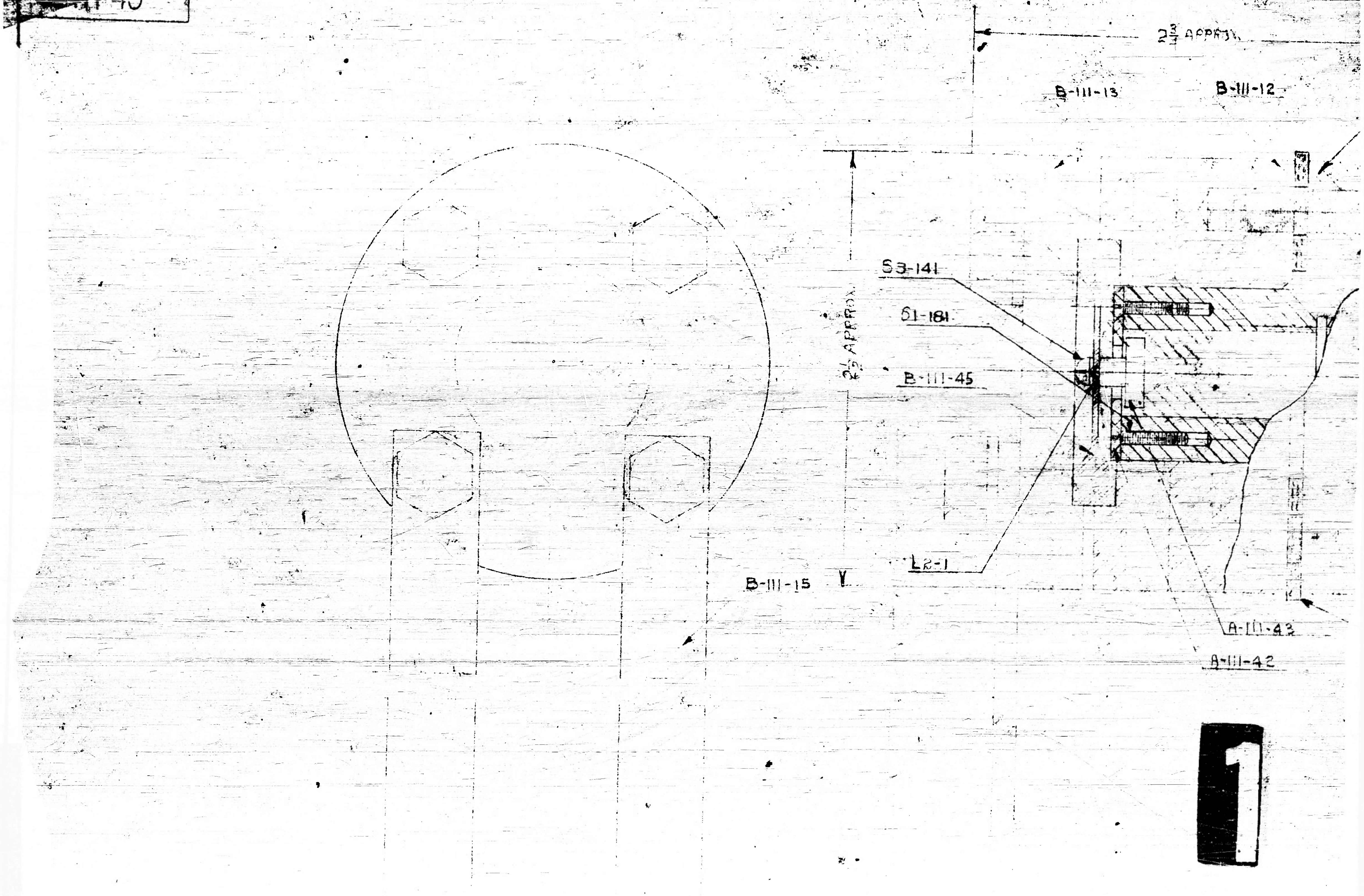
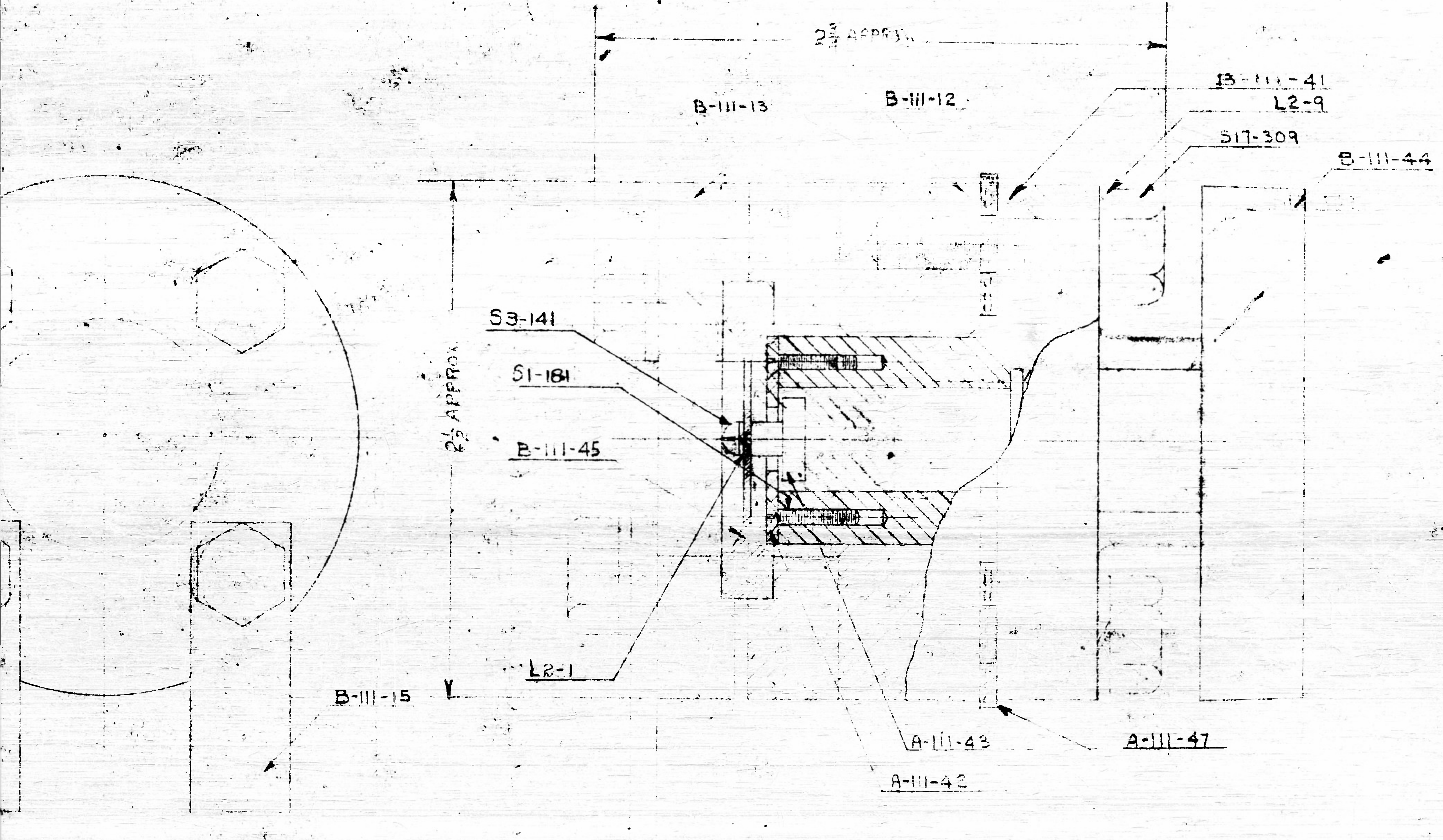


FIGURE 16

PRINTED BY SPAULDING-MOSS CO., BOSTON, MASS., RE-ORDER NO. A-1368


2 3/4 APPROX.
B-111-13
B-111-12
S3-141
S1-181
B-111-45
2 1/2 APPROX.
L2-1
B-111-15
A-111-43
A-111-42

1



2

PART NO.	DESCRIPTION	MAT'L	QTY.
B-111-41	VALVE BLOCK	[illegible]	1
B-111-12	[illegible]	ST.ST.	1
B-111-13	[illegible]	ST.ST.	1
B-111-45	[illegible]	[illegible]	1
B-111-15	[illegible]	[illegible]	1
S17-309	[illegible]	ST.ST.	8
L2-9	[illegible]	ST.ST.	8
A-111-42	VALVE SEAT	TOOL ST.	1
A-111-43	VALVE	TOOL ST.	1
B-111-44	ADAPTER	ST.	1

3

REVISIONS					
NO.	WAS	DATE	NO.	WAS	DATE
	D01/DR 936	[illegible]			

REDRAWN DATE WEIGHT

DR

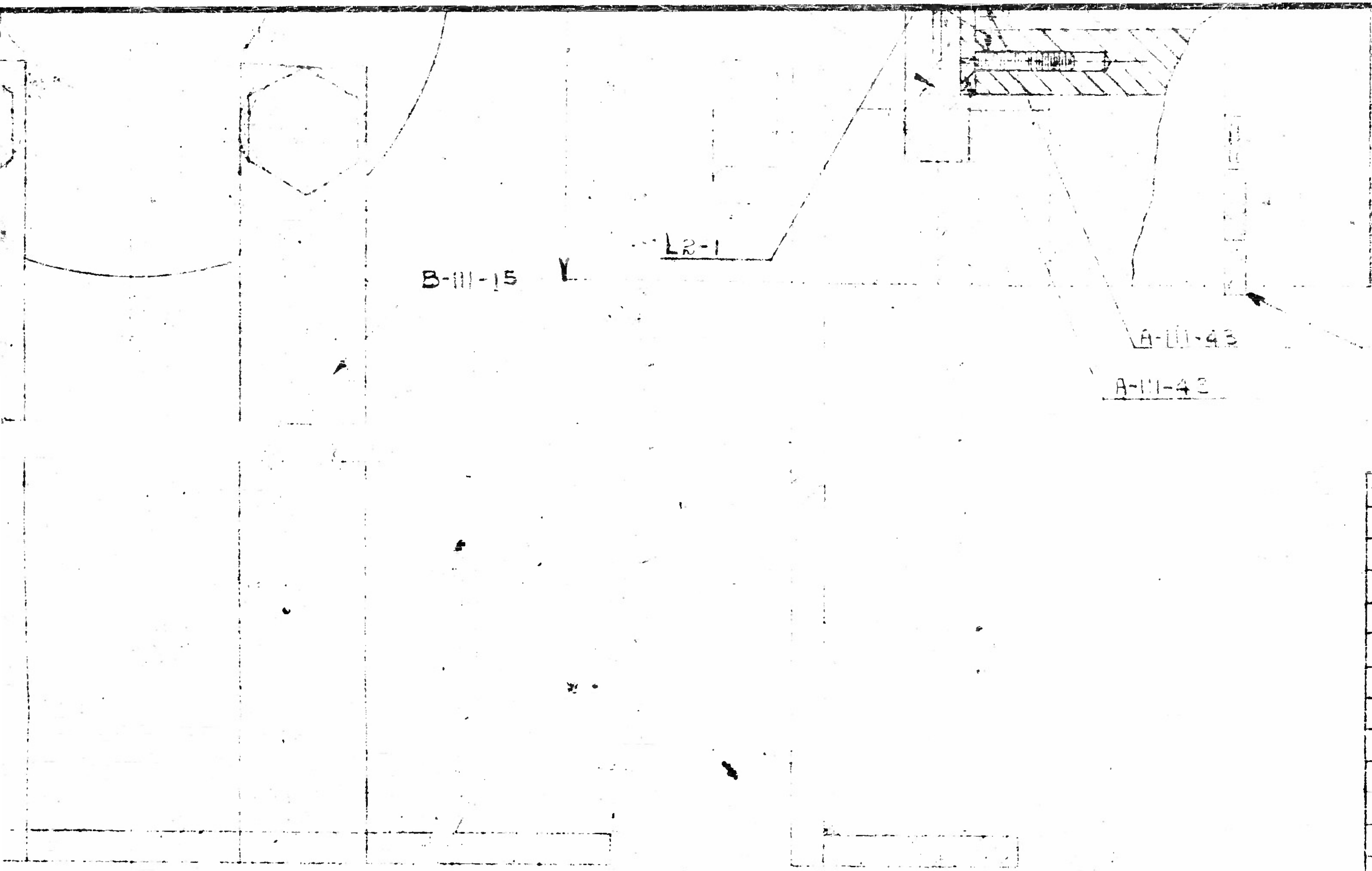
SCALE:

TW

FOR:

NAME:

UL



PART NO.	[illegible]	MAT'L	QTY.
B-111-41	VALVE BLOCK	ST.ST.	1
B-111-12	[illegible]	ST.ST.	1
B-111-13	[illegible]	ST.ST.	1
B-~~111~~-45	[illegible]	TOOL ST.	1
B-111-15	[illegible]	[illegible]	1
517-309	[illegible]	ST.ST.	8
L2-9	[illegible]	ST.ST.	3
A-111-42	VALVE SEAT	TOOL ST.	1
A-111-43	VALVE	TOOL ST.	1
B-111-44	ADAPTER	ST.ST.	1
S1-16.55	#2-56-N.C.-2X 3/8 LG FLAT HD MACH SCR	ST.ST.	4
S3-14155	#2-56-N.C.-2X 1/4 LG ROUND HD MACH SCR	ST.ST.	1
L2-1	#2 LOCKWASHER	ST.ST.	1
A-111-47	RUBBER SPACER	RBR.	1

ALL DIMENSIONS ARE IN INCHES UNLESS OTHERWISE SPECIFIED
TOLERANCE ON FRACTIONAL DIMENSIONS [illegible]
BREAK SHARP CORNERS UNLESS OTHERWISE SPECIFIED

REVISIONS

NO.	WAS	DATE	NO.	WAS	DATE
	D01/DR 936	[illegible]			

REDRAWN | DATE | WEIGHT

DR. [illegible] DATE [illegible] CKD [illegible] DATE [illegible] APPD [illegible] DATE [illegible]

SCALE: [illegible]

MATERIAL:

FINISH:

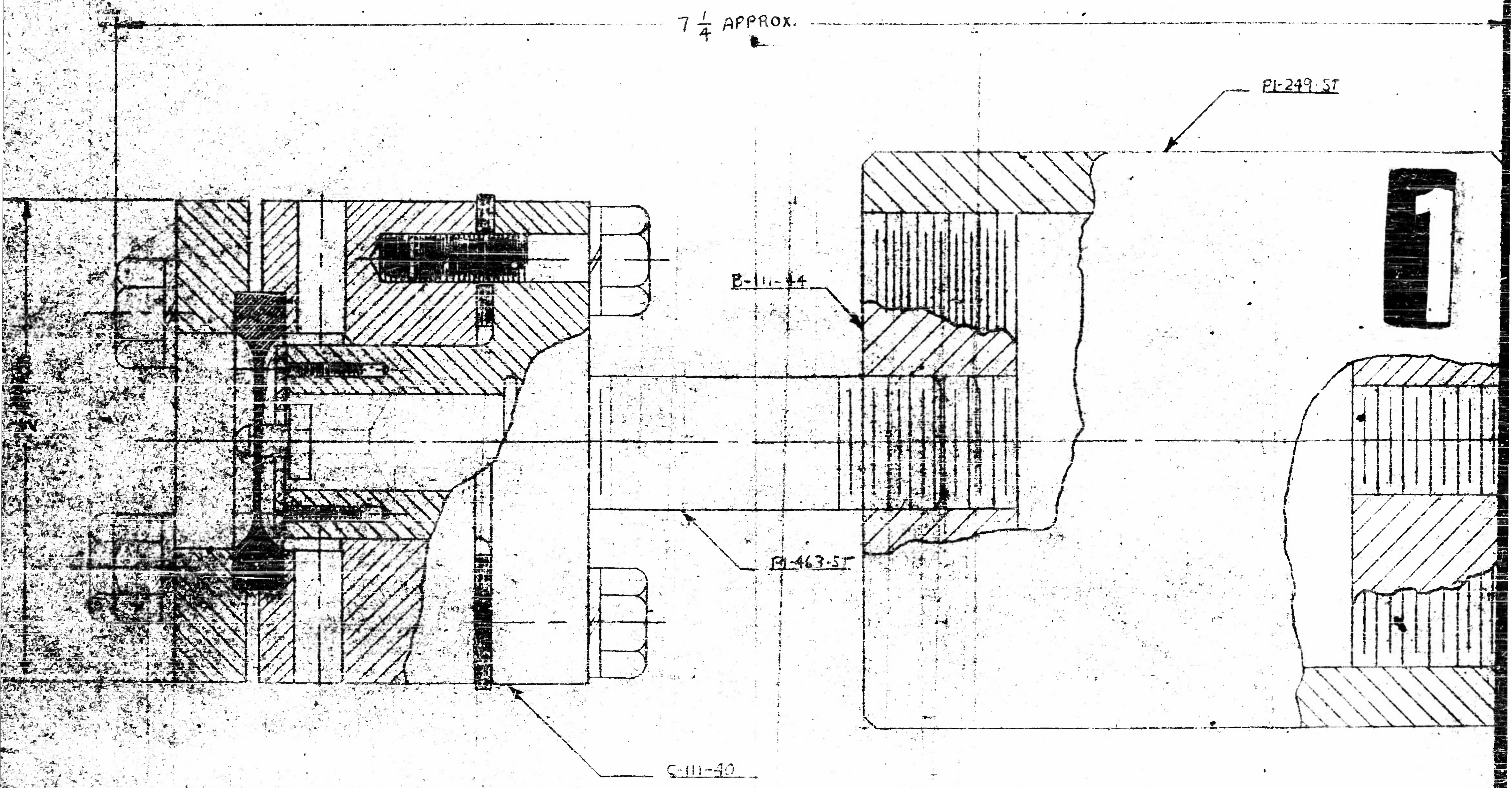
FOR: [illegible]

NAME SUB ASSY WATER HAMMER DEVICE

ULTRASONIC CORPORATION
CAMBRIDGE MASS.

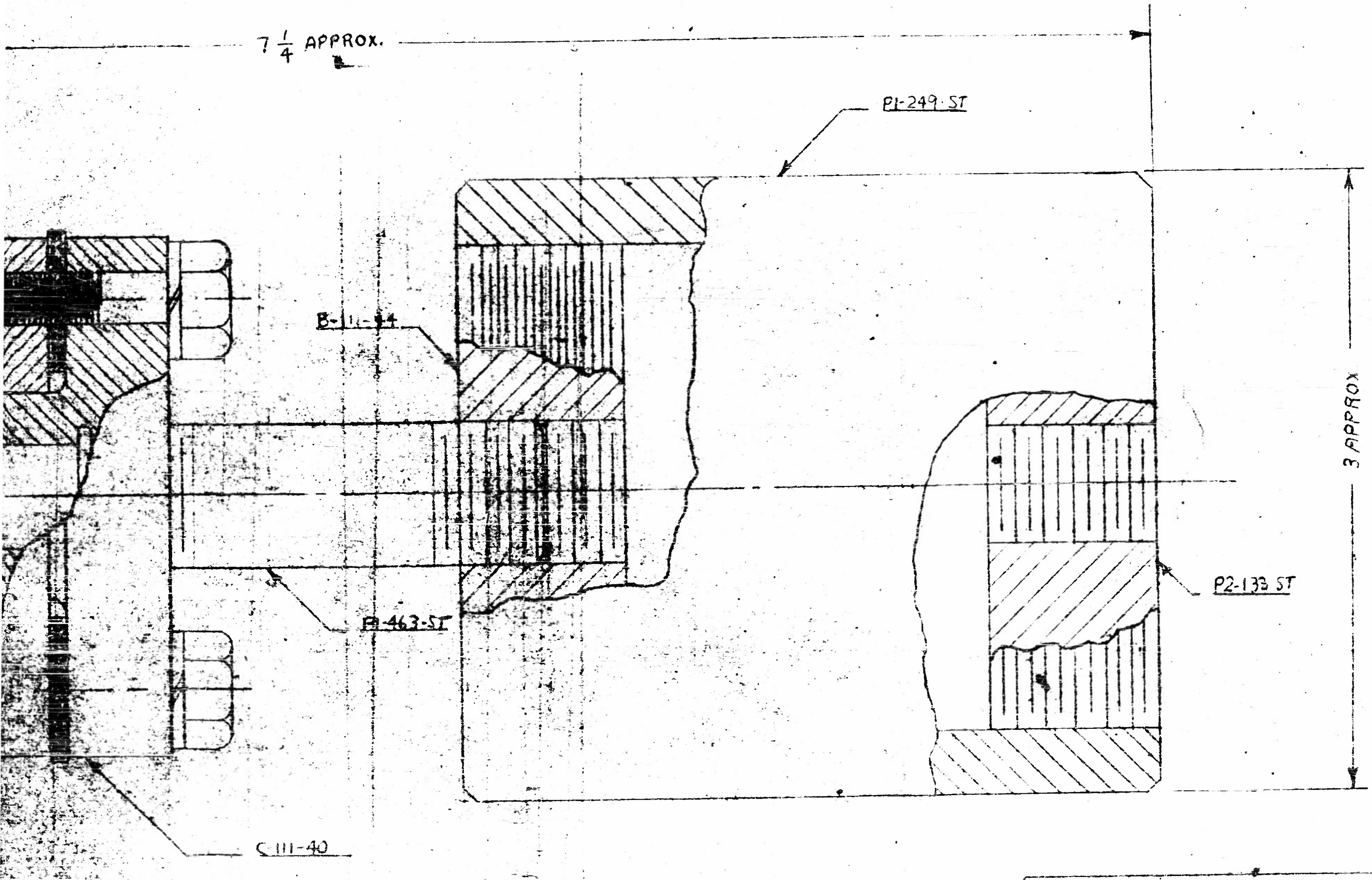
C-111-40

4



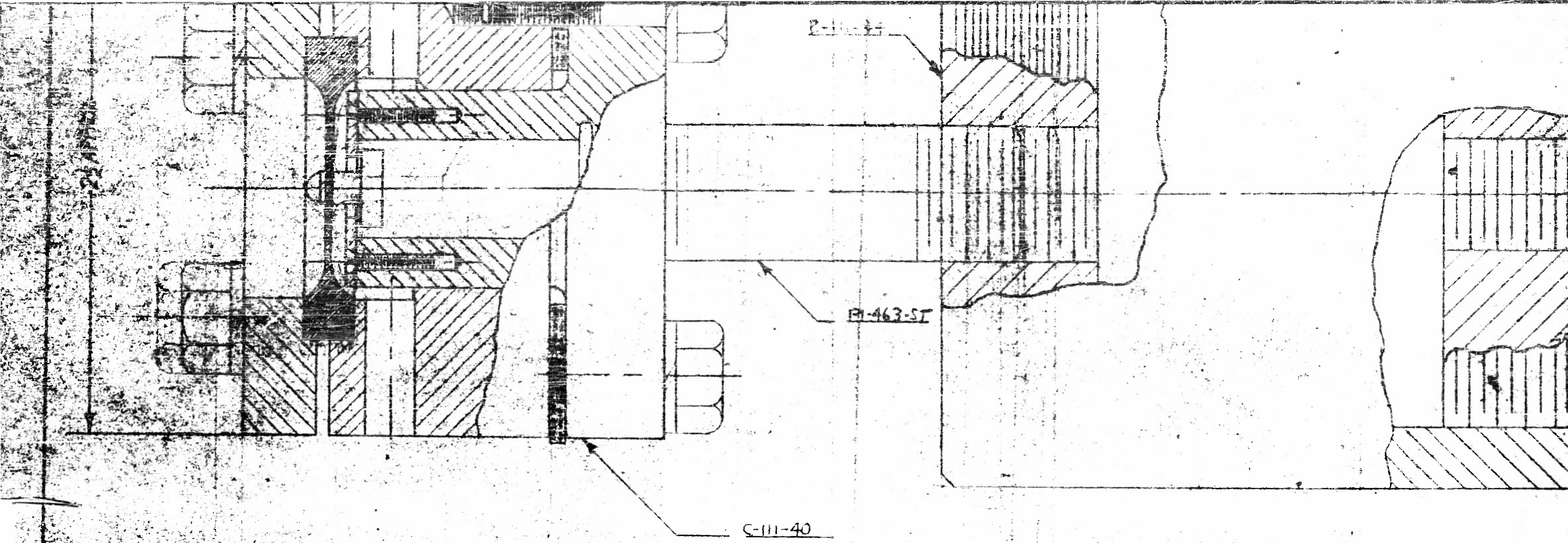
7 1/4 APPROX.
P1-249-ST
B-111-44
B-463-ST
C-111-40
1

REVISIONS			
SYM.	DESCRIPTION	DATE	APPROVAL



2

PARTS LIST



3

PART NO.
P1-249-ST
B-111-44
P2-132-ST
P1-463-ST
C-111-40

ALL DIMENSIONS ARE IN INCHES UNLESS OTHERWISE SPECIFIED

TOLERANCE ON FRACTIONAL DIMENSIONS ±1/64

TOLERANCE ON DECIMAL DIMENSIONS ±.005

TOLERANCE ON ANGULAR DIMENSIONS ±½°

BREAK SHARP CORNERS UNLESS OTHERWISE SPECIFIED

WEIGHT

SCALE:

USED ON

FOR

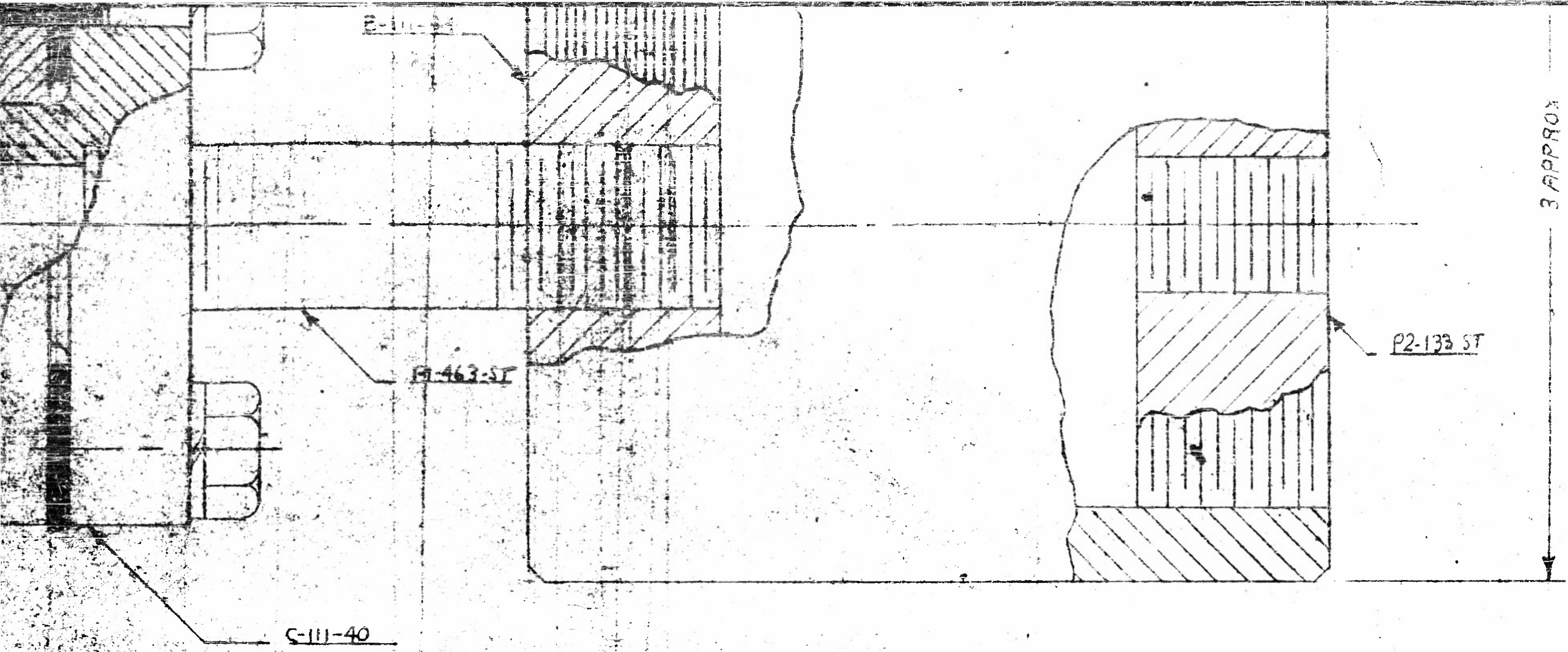
NAME

REVELATION OF THE CONTENTS OF THIS DRAWING TO AN UNAUTHORIZED PERSON IS PROHIBITED BY LAW.

SEE ESPIONAGE ACT 50 USC 31-32

DO NOT SCALE THIS DRAWING

ULTRASONIC



PARTS LIST			
PART NO.	DESCRIPTION	MAT'L	QUAN.
P1-249-ST.	PIPE CPLG. 2" EXTRA STRONG	STL.	1
B-111-44	ADAPTER	FST	1
P2-133-ST.	BUSH. REDUCING FLUSH 2 X ½	FST	1
P1-463-ST.	½ I.P.S.	STL.	1
C-111-40	SUB ASS'Y WATER HAMMER DEVICE		1

FIGURE 18

4

ALL DIMENSIONS ARE IN INCHES UNLESS OTHERWISE SPECIFIED TOLERANCE ON FRACTIONAL DIMENSIONS ± 1/64 TOLERANCE ON DECIMAL DIMENSIONS ± .005 TOLERANCE ON ANGULAR DIMENSIONS ± 1° BREAK SHARP CORNERS UNLESS OTHERWISE SPECIFIED	WEIGHT	DR. C 2 W DATE 8-26-52	CKD. DATE	APPD. DATE
	SCALE: 1" = 2"	MATERIAL:		FINISH:
USED ON	FOR: ONR 111			
	NAME: ASS'Y WATER HAMMER DEVICE			
DO NOT SCALE THIS DRAWING	ULTRASONIC CORPORATION CAMBRIDGE, MASS.			

REVELATION OF THE CONTENTS OF THIS DRAWING TO AN UNAUTHORIZED PERSON IS PROHIBITED BY LAW.

SEE ESPIONAGE ACT 50 USC 31-32

ULTRASONIC CORPORATION CAMBRIDGE, MASS.	TITLE: SCHEMATIC OF WATER HAMMER DEVICE	BY:	DATE: 8-24-52

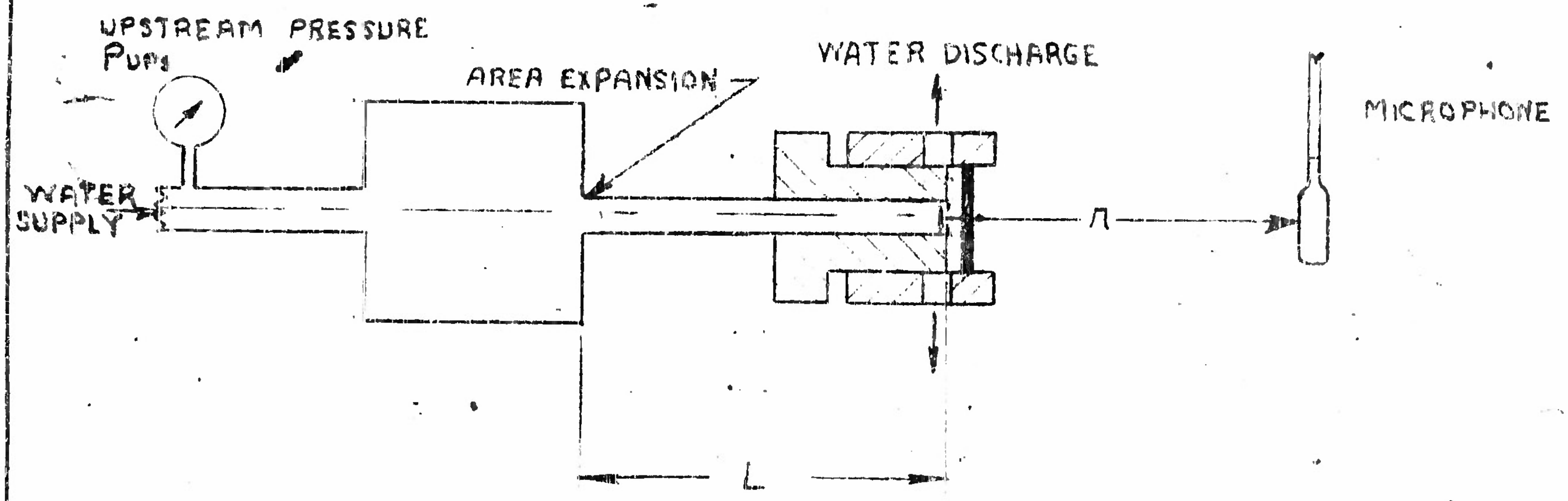
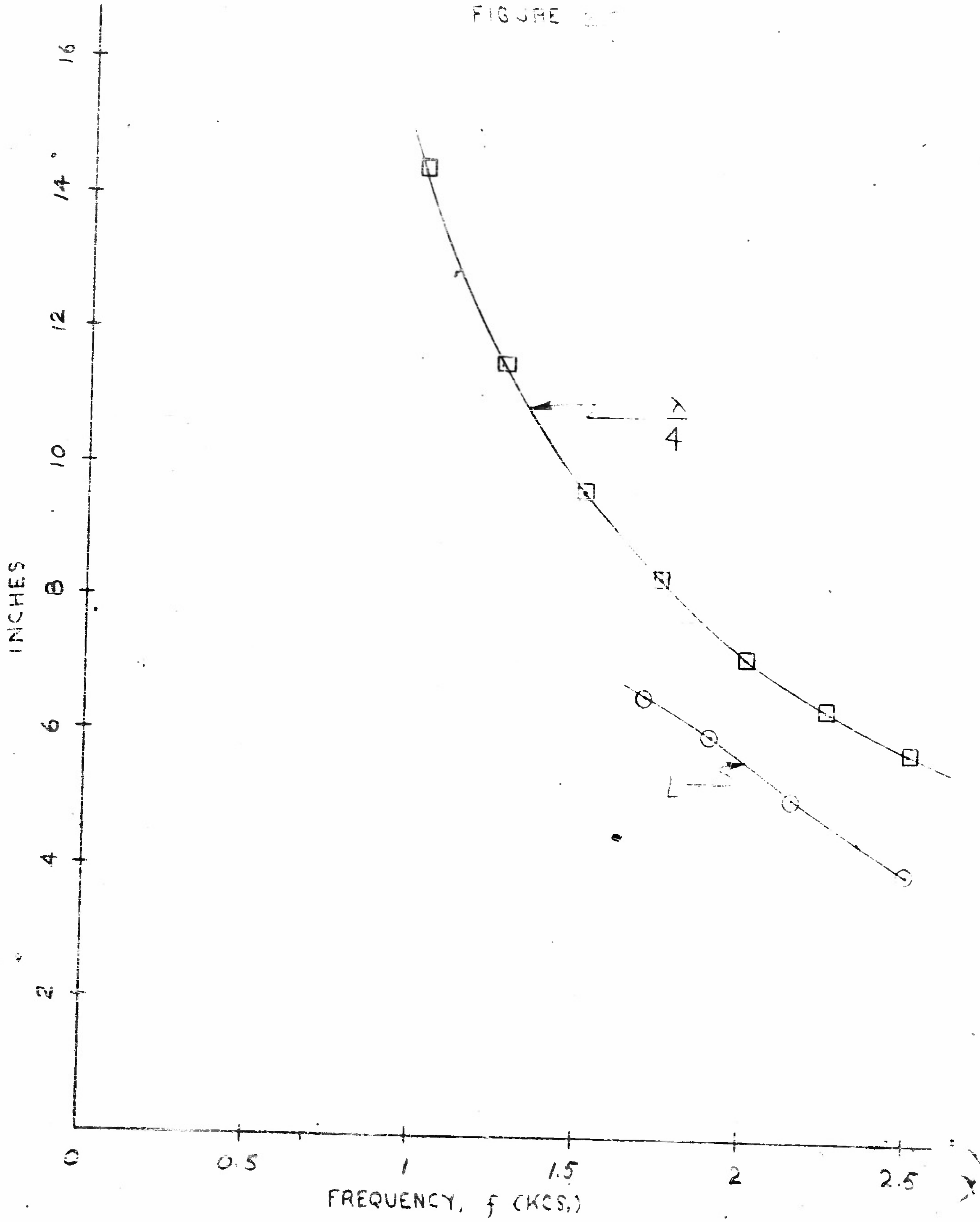


FIGURE 19

PRINTED BY SPAULDING-MOSS CO., BOSTON, MASS. RE-ORDER NO. A-1368

FREQUENCY VS. TUBE LENGTH



FIGURE

MODIFIED WATER HAMMER DEVICE ASSEMBLED FOR TESTING

FIGURE 21

PARTS OF MODIFIED WATER HAMMER DEVICE

CASTOR OIL CHAMBER

VALVE SEAT BLOCK AND VALVE SEAT

NOZZLE BLOCK

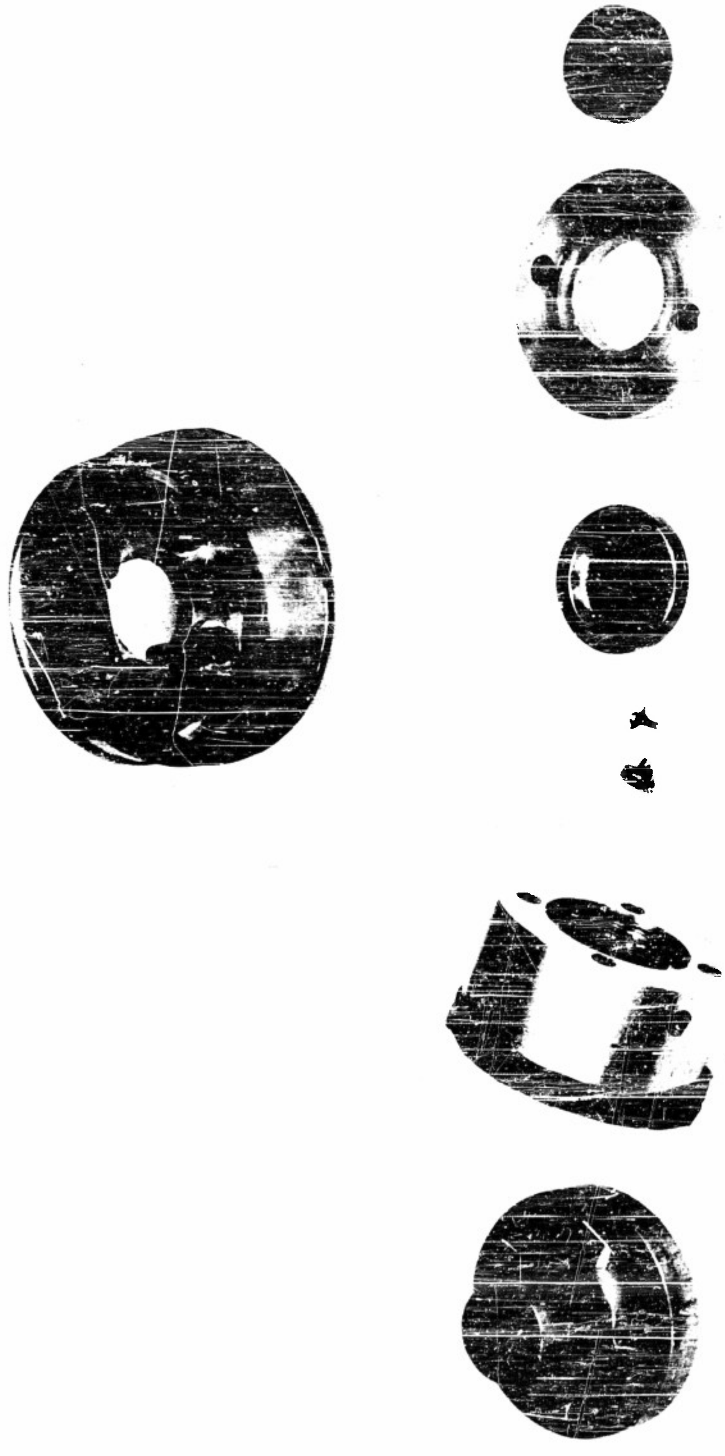
VALVE AND MOUNTING SCREWS

CLAMPING RING

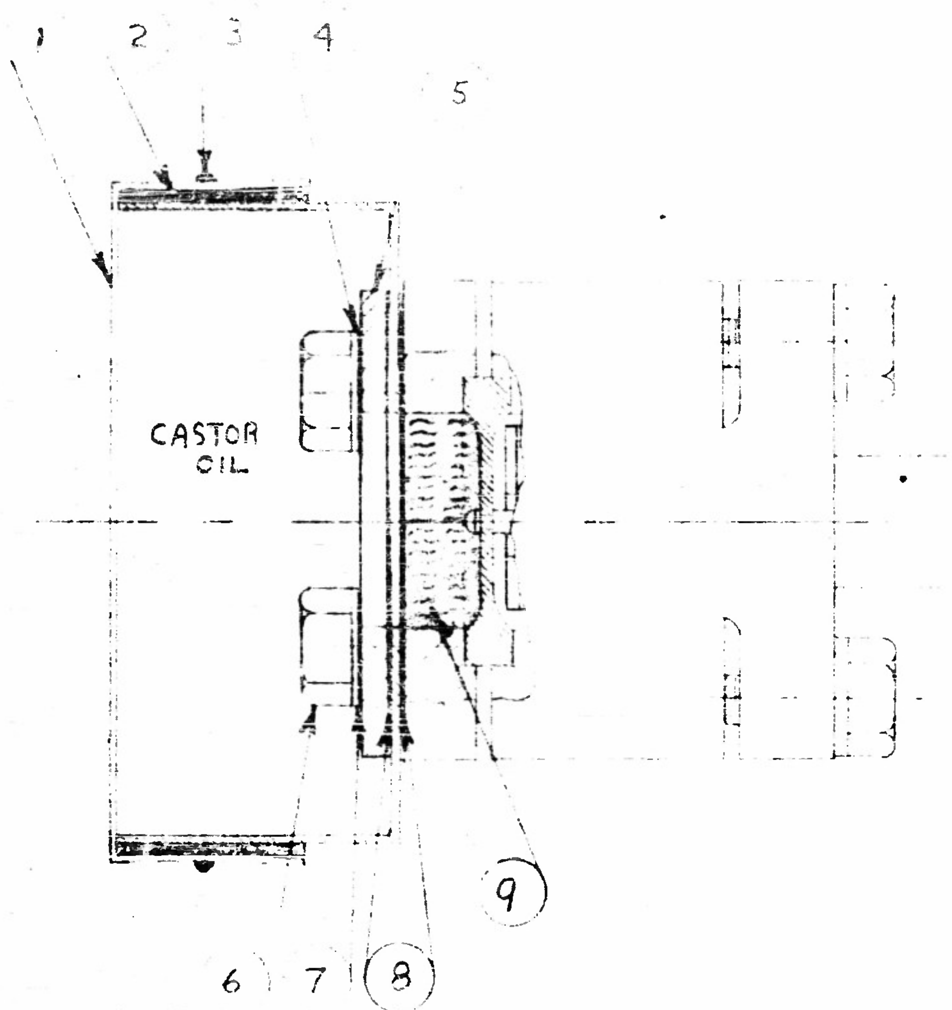
RUBBER SEALING PLUG

DIAPHRAGM

FIGURE 22



B-



PARTS LIST

ITEM	DESCRIPTION	MAT'L	QUAN
1	PLIOFILM MEMBRANE		1
2	FRICTION TAPE		
3	RUBBER BAND		
4	WASHER [illegible]	STL	4
5	CLAMPING RING	STL	1
6	[illegible] HEX HD 1/4 X 1 LG	STL	4
7	[illegible]		4
8	GASKET VELLUMOID		2
9	MOLDED RUBBER FILL		1

FIGURE 23

ALL DIMENSIONS ARE IN INCHES UNLESS OTHERWISE SPECIFIED
TOLERANCE ON FRACTIONAL DIMENSIONS ± 1/64
BREAK SHARP CORNERS UNLESS OTHERWISE SPECIFIED

REVISIONS					
NO	WAS	DATE	NO	WAS	DATE

REDRAWN DATE WEIGHT

DR. DATE CKD. DATE APPD. DATE

SCALE: FULL

MATERIAL:

FINISH:

FOR: [illegible]

NAME: MODIFIED WATER HAMMER DEVICE

ULTRASONIC CORPORATION
CAMBRIDGE, MASS.

B-

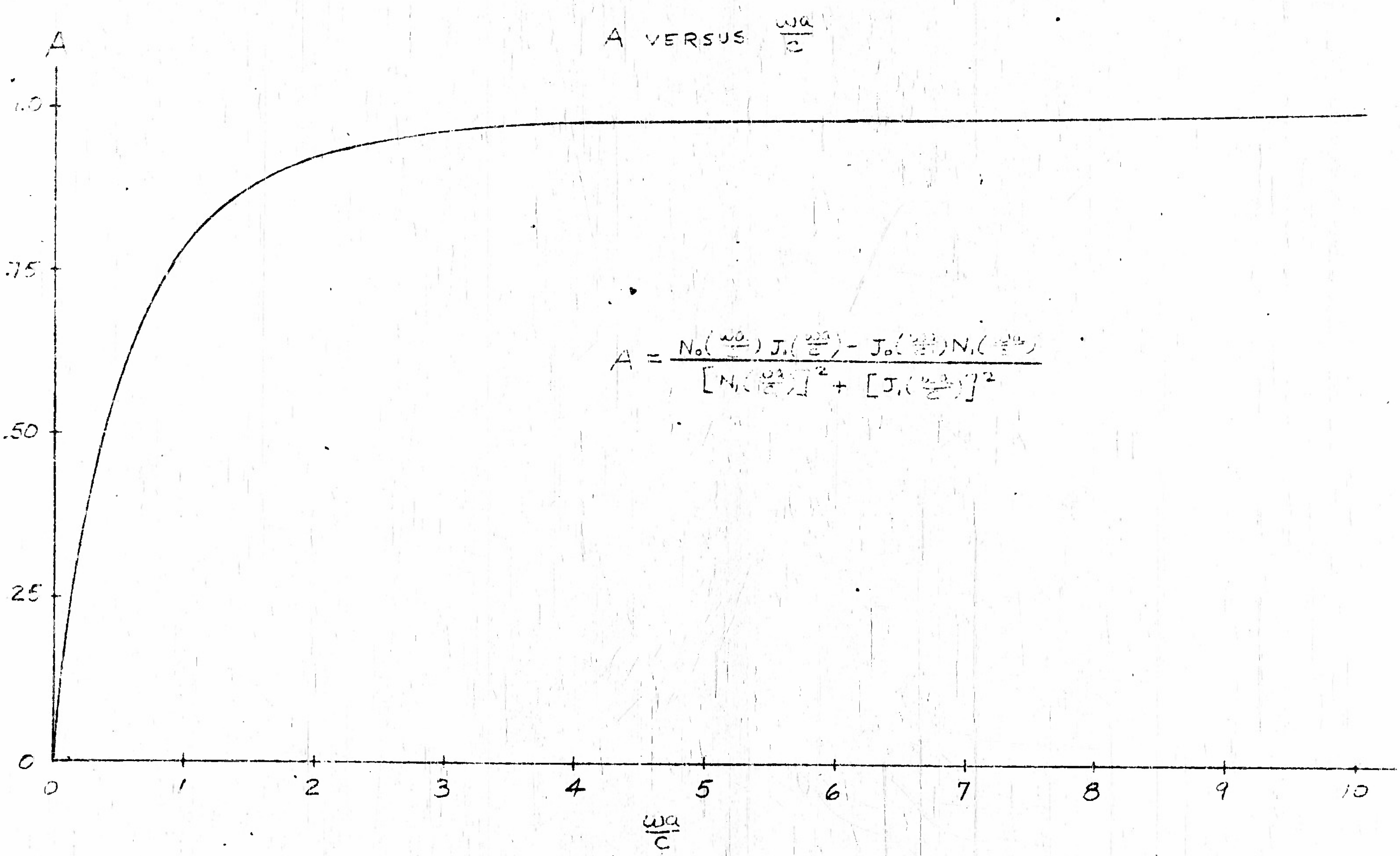

A VERSUS $\frac{\omega a}{c}$
A
1.0
.75
.50
.25
0
0
1
2
3
4
5
6
7
8
9
10
$\frac{\omega a}{c}$
$A = \frac{N_0(\frac{\omega a}{c})J_1(\frac{\omega a}{c}) - J_0(\frac{\omega a}{c})N_1(\frac{\omega a}{c})}{[N_1(\frac{\omega a}{c})]^2 + [J_1(\frac{\omega a}{c})]^2}$

FIGURE 24

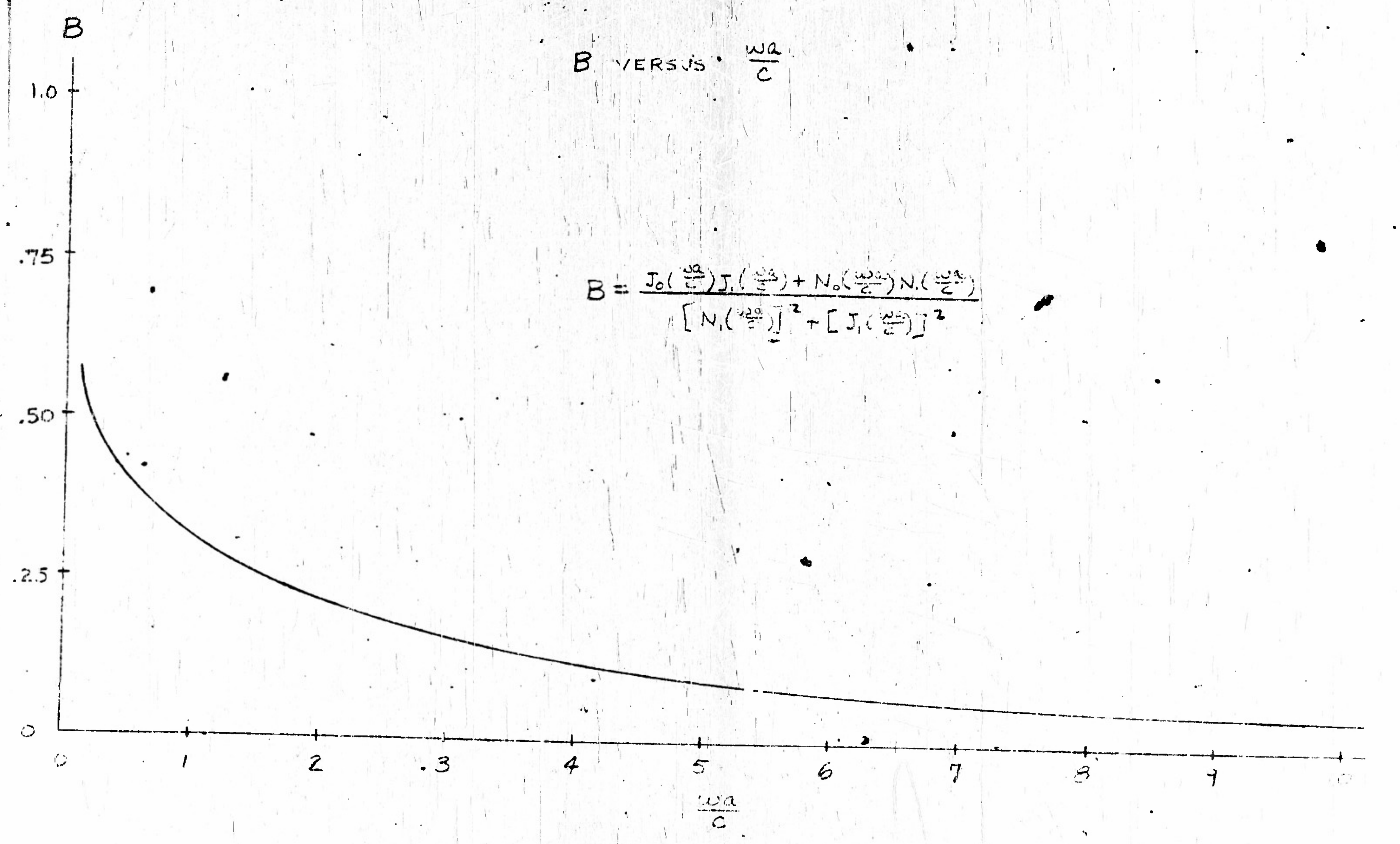

B
B VERSUS $\frac{\omega a}{c}$
$B = \frac{J_0(\frac{\omega a}{c})J_1(\frac{\omega a}{c}) + N_0(\frac{\omega a}{c})N_1(\frac{\omega a}{c})}{[N_1(\frac{\omega a}{c})]^2 + [J_1(\frac{\omega a}{c})]^2}$
1.0
.75
.50
.25
0
0
1
2
3
4
5
6
7
8
9
$\frac{\omega a}{c}$

FIGURE 25

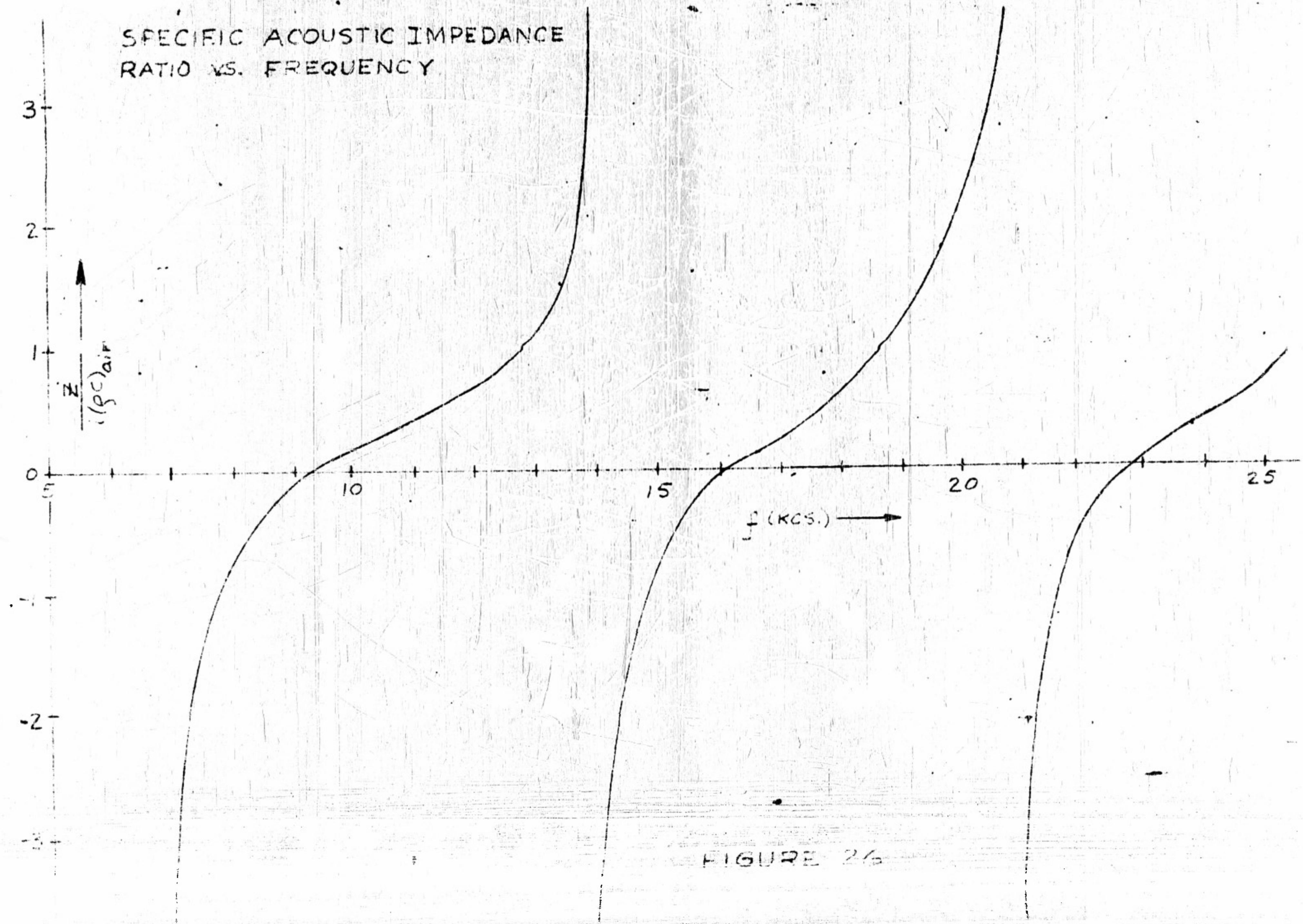

SPECIFIC ACOUSTIC IMPEDANCE
RATIO VS. FREQUENCY
3
2
1
0
-1
-2
-3
$\frac{Z}{(\rho c)_{air}}$
5
10
15
20
25
f (KCS.)

FIGURE 26